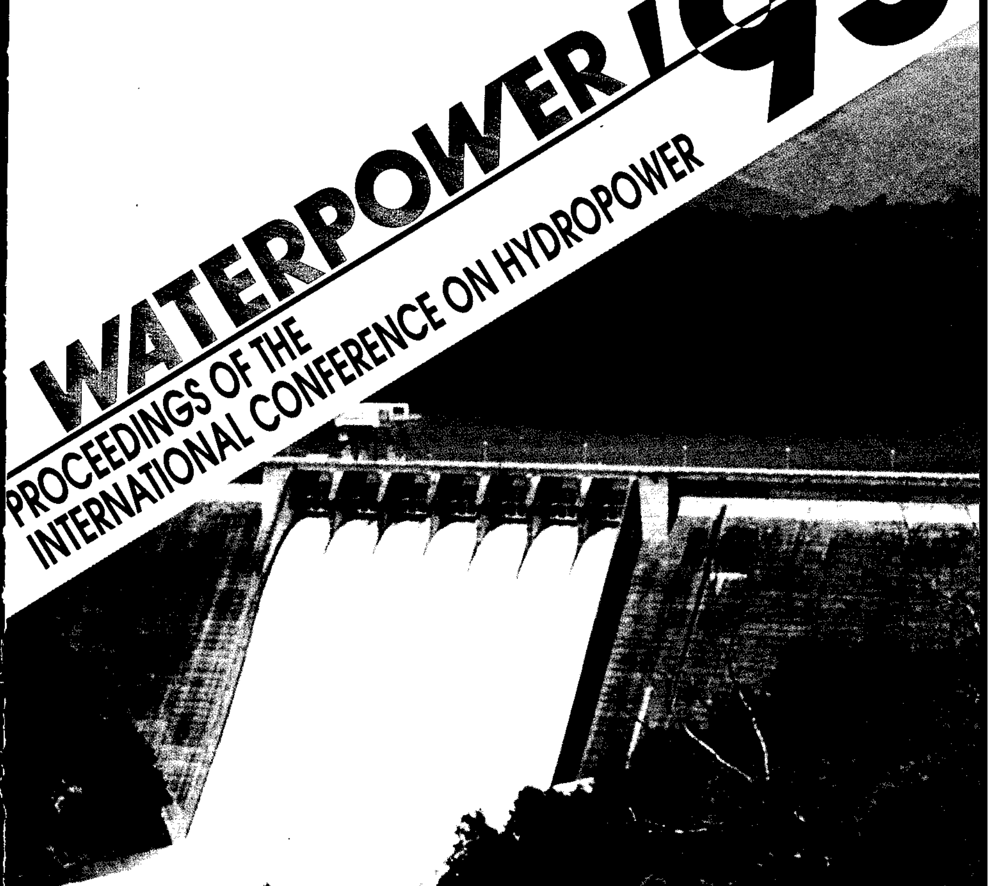
WATERPOWER

WATERPOWER '93

PROCEEDINGS OF THE INTERNATIONAL CONFERENCE ON HYDROPOWER



VOLUME 3

Edited by W. David Hall

WATERPOWER '93

PROCEEDINGS OF THE INTERNATIONAL CONFERENCE ON HYDROPOWER

Nashville, Tennessee
August 10-13, 1993

Edited by W. David Hall

Published by the
American Society of Civil Engineers
345 East 47th Street
New York, New York 10017-2398

Volume III

Rehabilitation & Modernization
Operations & Maintenance
Electrical Systems & Controls
Electrical & Generators
Turbines
Poster Forum

ABSTRACT

This proceedings contains the papers presented at the Waterpower '93 conference held in Nashville, Tennessee, August 10-13, 1993. The conference brought together owners, planners, engineers, regulators, manufacturers, and others to share vital information surrounding the conference theme, Hydropower - Its Role in World Energy. This eighth edition of the conference series which began in 1979 was hosted by the Tennessee Valley Authority and the U.S. Army Corps of Engineers, Nashville District. Subject areas include: 1) environmental issues; 2) legal factors; 3) planning; 4) hydraulics; 5) hydrology; 6) operation and maintenance; 7) rehabilitation; 8) research and development; 9) computer applications; 10) geotechnical, mechanical, and electrical systems; 11) reservoir system operation; and case works.

ISBN 0-87262-924-4
ISSN 1057-1841
Manufactured in the United States of America.

WATERPOWER '93 Nashville, Tennessee

TECHNICAL PROGRAM – OVERVIEW

Date	Day	Time	Carrol Room	Browning A Room	Browning B Room	Taylor B Room	Taylor A Room	Handy Room
August 10		5:30 - 8:30PM	Welcome Reception					
August 11	WEDNESDAY	8:30 - 10:00AM	OPENING PLENARY					
		10:00 - 10:45	break					
		10:45 - 12:15	ENVIRONMENTAL-1 Fish Passage Session 1	DAM SAFETY-1 Analysis Session 2	[illegible]	[illegible]	CASE STUDIES-1 Project Evaluation and Assessments Session 5	HYDRAULICS-1 Design of Hydraulic Structures Session 6
			lunch					
		1:45 - 3:15	ENVIRONMENTAL-2 Turbine Impacts on Fish Session 7	DAM SAFETY-2 Dam Rehabilitation Session 8	[illegible]	COMPUTER APPLICATIONS-1 (Pumped Storage & Project Systems) Session 10	CASE STUDIES-2 Innovations in Equipment and Design Session 11	HYDRAULICS-2 Modeling & Analysis Session 12
		4:00 - 5:30	ENVIRONMENTAL-3 Fish Protection Session 13	LICENSING/ LEGAL-1 Relicensing Issues Session 14	RESEARCH & DEVELOPMENT-1 Hydraulic Turbine Performance Session 15	[illegible]	CASE STUDIES-3 General-1 Session 17	[illegible]
August 12	THURSDAY	8:30 - 10:00	ENVIRONMENTAL-4 Regulatory Issues Session 19	DAM SAFETY-3 General Session 20	[illegible]	[illegible]	CASE STUDIES-4 Small Hydro Session 23	MECHANICAL SYSTEMS Session 24
		10:45 - 12:15	ENVIRONMENTAL-5 Instream Flows Session 25	DAM SAFETY-4 Instrumentation and Remote Sensing Session 26	ECONOMICS & FINANCE-1 Integrated Resources Analysis Session 27	RESEARCH & DEVELOPMENT-2 Project Planning & Operation Session 28	CASE STUDIES-5 Rehabilitation of Major Features Session 29	[illegible]
			lunch					
		1:45 - 3:15	ENVIRONMENTAL-6 (EPRI) Diversion Screens Session 31	LICENSING/ LEGAL-2 Licensing Issues Session 32	HYDROLOGY-1 Streams and Reservoirs Session 33	[illegible]	CIVIL WORKS-1 General-1 Session 35	[illegible]
		4:00 - 5:30	RESERVOIR/SYSTEM REGULATION Session 37	LICENSING/ LEGAL-3 General Session 38	[illegible]	[illegible]	CIVIL WORKS-2 General-2 Session 41	ECONOMICS & FINANCE-2 Case Study Session 42
			Conference Dinner					
August 13	FRIDAY	8:30 - 10:00	ENVIRONMENTAL-7 Water Quality Session 43	DAM SAFETY-5 (EPRI) Studies and Models Session 44	[illegible]	COMPUTER APPLICATIONS-2 Control & Efficiency Session 46	GEOTECHNICAL-1 Studies for Dams Session 47	[illegible]
		10:45 - 12:15	ENVIRONMENTAL-8 Case Studies Session 49	PLANNING Session 50	HYDROLOGY-2 Flood and Storm Prediction Session 51	[illegible]	GEOTECHNICAL-2 Studies for Project Features Session 53	CASE STUDIES-6 General-2 Session 54
			Closing Plenary					

○ Volume 1 ◍ Volume 2 ● Volume 3 *(includes posters)*

CONTENTS

VOLUME I

Planning
Chair: Richard L. Mittelstadt, US Army Corps Of Engineers, Portland, OR
Session 50

Environmental-1: Fish Passage
Chair: James E. Crews; HQ, US Army Corps Of Engineers; Washington, DC
Session 1

Environmental-2: Turbine Impacts On Fish
Chair: Rudd Turner, CENPE Environmental Resources Division, Portland, OR
Session 7

Environmental-3: Fish Protection
Chair: Richard Armstrong; HQ, US Army Corps Of Engineers
Session 13

Environmental-4: Regulatory Issues
Chair: Richard McDonald, CEWRC-WLRC, Portland, OR
Session 19

Environmental-5: Instream Flows
Chair: Art Denys, US Army Corps Of Engineers, Dallas, TX
Session 25

Environmental-6 (EPRI): Diversion Screens
Chair: Charles Sullivan, Electric Power Research Institute, Palo Alto, CA
Session 31

Environmental-7: Water Quality
Chair: Ralph Brooks, TVA Water Resources Division, Knoxville, TN
Session 43

Environmental-8: Case Studies
Chair: James R. Hanchey, US Army Corps Of Engineers, Vicksburg, MS
Session 49

Licensing/Legal-1: Relicensing Issues
Chair: Louis Rosenman, Attorney At Law, Washington, DC
Session 14

Licensing/Legal-2: Licensing Issues
Chair: Fred E. Springer, Federal Energy Regulatory Comm., Washington, DC
Session 32

Licensing/Legal-3: General
Chair: Thomas Russo, Federal Energy Regulatory Commission, Washington, DC
Session 38

Case Studies-1: Project Evaluation And Assessments
Chair: Edward F. Carter, Harza Engineering Co., Chicago, IL
Session 5

Case Studies-2: Innovations in Equipment and Design
Chair: Arvids Zagars, Harza Engineering Co., Chicago, IL
Session 11

Case Studies-3: General-1
Chair: Foster Pelton, Acres International Corp., Amherst, NY
Session 17

Case Studies-4: Small Hydro
Chair: Ashok K. Rajpal, Mead & Hunt, Inc., Madison, WI
Session 23

Case Studies-5: Rehabilitation Of Major Features
Chair: Richard D. Stutsman, Pacific Gas & Electric Co., San Francisco, CA
Session 29

Case Studies-6: General-2
Chair: Edgar T. Moore, Consulting Civil Engineer, Naperville, IL
Session 54

VOLUME II

Reservoir/System Regulation
Chair: Ron Yates, US Army Corps Of Engineers, Cincinnati, OH
Session 37

Dam Safety-1: Analysis
Chair: Joseph L. Ehasz, Ebasco Services Inc., New York, NY
Session 2

Dam Safety-2: Dam Rehabilitation
Chair: Alton P. Davis, GEI Consultants Inc., Winchester, MA
Session 8

Dam Safety-3: General
Chair: Jerrold W. Gotzmer, Federal Energy Regulatory Comm., Washington, DC
Session 20

Dam Safety-4: Instrumentation And Remote Sensing
Chair: Daniel J. Mahoney, Federal Energy Regulatory Comm., Washington, DC
Session 26

Dam Safety-5 (EPRI): Studies And Models
Chair: Douglas I. Morris, Electric Power Research Institute, Palo Alto, CA
Session 44

Research And Development-1: Hydraulic Turbine Performance
Chair: John J. Cassidy, Bechtel Corp., San Francisco, CA
Session 15

Research And Development-2: Project Planning And Operation
Chair: Frederick A. Locher, Bechtel Corp., San Francisco, CA
Session 28

Hydrology-1: Streams And Reservoirs
Chair: Andar Lin, New York Power Authority, White Plains, NY
Session 33

Hydrology-2: Flood And Storm Prediction
Chair: Catalino Cecilio, Consulting Engineer, San Francisco, CA
Session 51

Computer Applications-1: Pumped Storage And Project Systems
Chair: John Bessaw, HDR Engineering, Boise, ID
Session 10

Computer Applications-2: Control & Efficiency
Chair: Jose Tello, Pacific Gas & Electric Co., San Ramon, CA
Session 46

Civil Works-1: General-1
Chair: Hugh G. McKay, Duke Power Company, Charlotte, NC
Session 35

Civil Works-2: General-2
Chair: David Burgoine, Acres International Corp., Amherst, NY
Session 41

Geotechnical-1: Studies For Dams
Chair: David E. Kleiner, Harza Engineering Co., Chicago, IL
Session 47

Geotechnical-2: Studies For Project Features
Chair: J. Lawrence Von Thun, US Bureau Of Reclamation, Denver, CO
Session 53

Hydraulics-1: Design Of Hydraulic Structures
Chair: George E. Hecker, Alden Research Laboratories, Holden, MA
Session 6

Hydraulics-2: Modeling And Analysis
Chair: Robert W. Kwiatkowski, Parsons Main, Inc., Boston, MA
Session 12

Mechanical Systems
Chair: Divu Narayan, New York Power Authority, White Plains, NY
Session 24

VOLUME III

Rehabilitation & Modernization-1: General-1
Chair: Thomas W. Plunkett, US Army Corps Of Engineers, Dallas, TX
Session 3

Rehabilitation & Modernization-2: General-2
Chair: Doug Filer, US Army Corps Of Engineers, Portland, OR
Session 9

Rehabilitation & Modernization-3: Redevelopment-1
Chair: Dick Hunt, Richard Hunt Assoc., Annapolis, MD
Session 21

Rehabilitation & Modernization-4: Pumped Storage
Chair: Bob Kohne, Pacific Gas & Electric Co., San Francisco, CA
Session 39

Rehabilitation & Modernization-5: Redevelopment-2
Chair: Duke Loney, US Army Corps Of Engineers, Portland, OR
Session 45

Operations & Maintenance-1: General Maintenance-1
Chair: Ron Loose, Us Department Of Energy, Washington, DC
Session 4

Electrical & Generators
Chair: Ron Domer, Consulting Engineer, San Francisco, CA
Session 40

Turbines-1: Design
Chair: Jody Key, Bonneville Power Administration, Portland, OR
Session 18

Turbines-2: Manufacturing
Chair: Allen Lewey, US Army Corps Of Engineers, Portland, OR
Session 30

Turbines-3: Testing
Chair: Rodney Whittinger, US Army Corps Of Engineers, Portland, OR
Session 36

Turbines-4: Pump Storage
Chair: Rochell Bates, Bonneville Power Administration, Portland, OR
Session 48

Poster Forum
Chair: Arlo Allen, US Bureau Of Reclamation, Salt Lake City, UT
Session 55

CONCRETE PENSTOCK LEAKAGE, INVESTIGATION OF CAUSES AND REHABILITATION OPTIONS

Param D. Bhat [1] Ph.D., P. Eng., M.ASCE

ABSTRACT:

Ontario Hydro has a number of hydroelectric generating stations where the steel lined concrete penstocks are connected to the main dam through an unlined concrete transition section. At many of these transition sections, severe water leakage has been occurring, particularly through construction joints, resulting in concrete deterioration and causing safety concerns for the structures and facilities downstream.

This paper describes the investigations undertaken to study the above problems in ten generating stations involving a total of 35 penstocks. The factors which contributed to the problem have been identified, possible remedial measures were investigated and solutions suggested based on the severity of the problem at each station.

1.0 INTRODUCTION:

A large number of hydroelectric generating stations (HGS) in the Ontario Hydro's generating system have reinforced concrete penstocks where each penstock is made of two segments: a long circular segment made of steel lined concrete section and a concrete transition section of varying geometry which connects the circular penstock to the dam through the outlet gates. A typical layout of such a penstock is shown in Figure 1.

[1] *Design Engineer Specialist, Hydroelectric Engineering and Construction Services, Ontario Hydro, 700 University Avenue, Toronto, Ontario, M5G 1X6.*

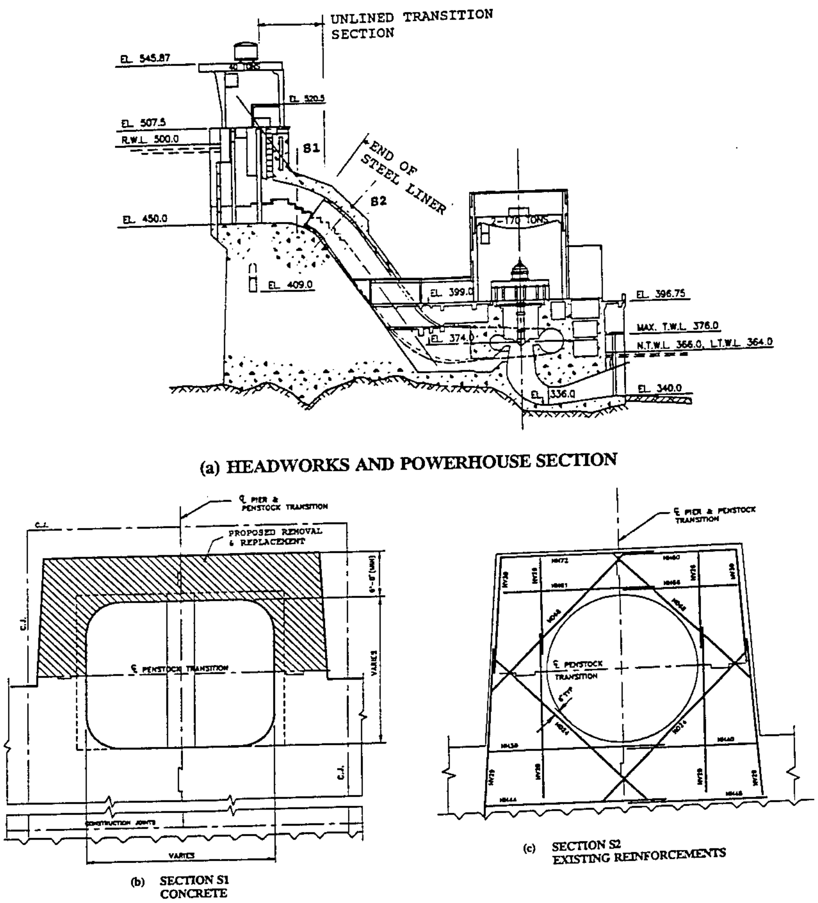


FIGURE 1: DES JOACHIMS G.S. - PENSTOCK LAYOUT

At a number of stations with the above general features, there has been leakage of water through construction joints and cracks in the concrete resulting in progressive deterioration and ice build up during winter months causing serious safety concerns to the power house roof and transformers on the roof decks. Repair of the power house roof has also been difficult due to the constant flow of water.

To address these concerns a detailed study of the problems were initiated to determine the causes and to identify possible remedial measures. The

transitionsections at ten generatingstations with a total of 35 penstocks were identifiedas the candidatesto be studied. The scope of the work undertaken are:

1. Field inspection and collection of all data available,
2. Determination of the cause of the problem for each location. This involved assessment of the original design and construction details, and
3. Generation of alternatives for a long-term solution to the leakage problem.

2.0 COMMON STRUCTURAL FEATURES:

Among the stations under investigationthe oldest was built in 1942 and the latest was built in 1970. The diameter of the penstocks varied from 12 feet to 25 feet and the length of the unlined segments varied from 2feet to 55 feet. All the transition sections studied had many common features. With the exception of one all were surface penstocks exposed to the atmosphere on three sides. In one station the penstocks were in a rock cut and only the top was exposed. They were all designed using the same basic design approaches namely the concrete sections were designed to withstand all the forces and provide leak tightness. However, differences exist in the configuration and details of construction joints and details of steel liner/concrete interface.

Some other salient features of interest are:

- penstocks in stations built prior to 1958 have no special provisions for sealing construction joints and the steel penstock/concrete interface.

- prior to 1963 a "step joint" was used on the side walls of the transition section. Effect of joints were not considered during the design.

- The steel linerfrom penstocks were terminatedat the transitionsection and were not extended up to the head gate, leaving varying lengths of unlined concrete. The excessive cost of steel at the time and the complicated geometry involved in fabricatingare suspected to be the reasons for not lining the concrete.

Stations built after 1958 incorporate water stops at construction joints and a special sealing detail at liner/concrete junction. Steel drains were also provided. The leakage and concrete deterioration are quite severe at 5 stations. At others cracks and leakage are minor.

Several types of repairs have been attempted in the past. They include: a) Replacing existing sealant in step joints with polysulphide sealants, chipping

and sealing of cracks and b) injection of various types of grouts in to the joints and cracks. This type of repair has been done both from inside and outside. Types of grouts used include cement grouts, chrome lignin grout, epoxy and urethanes. These repairs had limited or short term success. In some cases grouting has resulted in plugging of the drains, thus increeasing the leakge and causing additional problems.

3.0 INVESTIGATIONS CARRIED OUT:

To achieve the objectives indicated earlier the following investigations were carried out for each of the stations:

- Compiling of all available information: design notes, drawings, specifications, construction records and repair history,
- Identification of the present day Design Requirements,
- Analytical review of the original design using the present day design requirements plus any special investigations e.g: thermal loads, pore pressure.
- Site inspection and special investigations such as bore hole observations.

In the following section, details of the investigations and findings at one of the severely deteriorated penstocks at Des Joachims (pronounced "Da Swisha") Generating Station are reported. Other penstocks have lesser deterioration and their repair options are discussed later.

4.0 DES JOACHIMS HGS:

4.1 General:

This station is about 40 years old and has 8 penstocks of 22 feet dia. each. The unlined transition section is about 18 feet long. The unlined section and about 45 feet of the steel penstock below it have a concrete wall thickness of about 6 to 9 feet. This section is referred to as the "transition block". Below this block the steel penstocks are encased in about 2 feet thick reinforced concrete "envelope". Only a small length of this envelope is exposed and the remaining length is within the powerhouse building (see Figure 1 (a)).

The side walls of the transition section have a longitudinal construction joint, at about mid height, which is formed in a number of steps. The vertical construction joint is at about the center of the transition section. No water stops have been used at joints. An emulsified asphalt joint sealant has been used on the inside face of the joints.

Drawings show one small peripheral sand drain around the steel penstock which is connected to a longitudinal drain running down the length of the penstock.

4.2 ANALYTICAL REVIEW:

DESIGN LOADS: The design loads considered in the original design were: static hydraulic load, pressure rise due to sudden closure, static hydraulic thrust due to change in direction, dynamic hydraulic thrust due to centrifugal force and weight of concrete.

Load effects not considered in the original design are: temperature loads, creep and shrinkage, pore pressure from water in the construction joints, seismic loads, snow, ice, wind loads and the effect of main dam deflection/movement.

Effect of construction joints were not explicitly considered in the design (e.g: effect of pore pressure at the joint). Calculations show that if water enters the joint due to poor bond etc., it will exert enough pressure to keep the joint open.

Most of the stations studied are located in northern parts of the province of Ontario where temperature fluctuations are quite significant. The original design did not consider temperature effects. In this review the effects of temperature were reviewed to determine the influence of temperature gradient between inside and outside of the section and effect of seasonal variation in temperature and possible effect of steel penstock expansion.

The review and analysis indicates that the thermal gradients across the depth of the section contribute to the development of tensile forces and widening of cracks. While it should be included in the design, it is not expected to be a major factor. The effect of seasonal variation and the expansion of steel penstock are not significant and can be neglectd in the design.

DESIGN METHOD: The roof and walls of the transition section were designed as fixed end beams. The end shear reactions were assumed to be the "bursting force" and the reinforcementon the inside of the wall were designed to withstand these bursting forces. The outside steel in the wall and roof "beams" were designed to take bending from hydrostatic loads. The design approach used appears to be satisfactory from strength point of view but lacks consideration of the leak tightness requirement of the penstock. However, for present designs, because of the availablity of computers a detailed analysis of the transition section shall be carried out considering this section as a part of the main dam. By this approach the influence of the main dam movement can also be taken in to account.

ALLOWABLE STRESSES: The design assumed that the smooth round reinforcing bars used have an allowable stress of 18000 psi. This assumed value appears to be high since, the general practice at the time was to use 12000 psi. allowable for water retaining structures. The walls have a reinforcement of about 0.43% transverse and 0.07% longitudinal.

REBAR SPLICE AND BOND STRESS: Another aspect observed in the review was that all transverse rebars were lapped at the stepped joint without staggering (see Figure 1(c)). Although adequate lap length (54 d for plain bars) was provided, the calculated bond stresses in some cases exceeded the assumed allowable (100 psi). However, they were taken to be adequate due to the confining effect provided by the steel in the normal direction.

The review indicates that although there are no major mistakes in the original design there are a number of shortcomings. Lack of explicit consideration of leak tightness requirements in the design is a major factor. If the assumptions are modified to include additional considerations discussed above, it is apparent that the rebars provided are inadequate and they would be stressed beyond their allowable stress. Also, the design lacked corner bars at stepped joints.

4.3 SITE INVESTIGATIONS:

The penstocks at this station had leakage problems almost from the start. Leakage exists through all seasons, however, the effect is more severe in the winter since the ice which accumulates on the penstock outside breaks off in large pieces which fall on to the powerhouse roof causing damage.

Water leakage does not exist when the unit is shutdown at the top gate and the penstock is dewatered, indicating that the leakage does not come from the main dam and originates downstream of the gates.

4.3.1 EXTERNAL VISUAL INSPECTION:

External inspection indicated extensive cracking on all surfaces, leakage, freeze-thaw damage and rust stains. On the side of two penstocks water spouts were visible inaddition to numerous active slow leaks. Leakage through the roof was minor.

The cracking, concrete deterioration and water leakage patterns were similar in all eight penstocks. In all units a horizontal crack extended from one wall stepped joint horizontally across the face to the joint on the other side wall.

Deterioration was quite severe in the steel penstock envelope concrete as well.

4.3.2 INTERIOR VISUAL INSPECTION:

Visual inspection of the inside of the transition section was also done when the penstock was empty, using a swing stage. However, the inspection had to be done from a distance due to safety reasons. There were no visible signs of distress or possible leak locations in the concrete surface or the joints.

4.3.3 BORE HOLE INVESTIGATION:

To investigate the condition at the stepped joint in as much detail as possible a bore hole was drilled from the powerhouse roof into the concrete wall of the transition block at unit 6 at an angle of 37 degrees to the horizontal (see Figure 2). The bore hole was parallel to the centerline of the penstock and intercepted the stepped joint at several places. The condition of the 1.75" diameter core sample from the bore hole up to about 40 feet in length was documented in detail. The condition of the concrete in this core was quite good except at the joint locations.

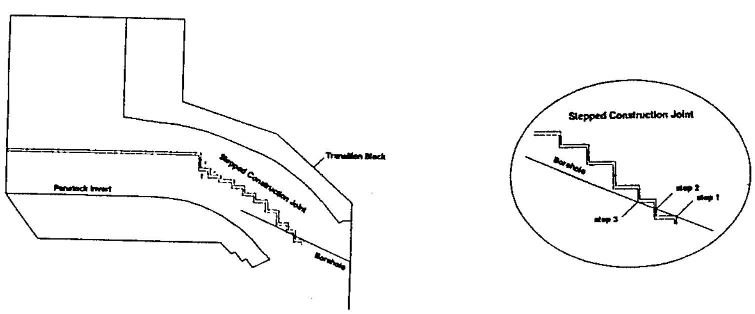


FIGURE 2: BORE HOLE LOCATION

A closed circuit television (CCTV) survey was conducted in the borehole. Gaps noted in the horizontal and vertical planes in three of the steps of the joint were measured. Erosion of the concrete in these steps was evident. The water leakage coming through the borehole was about 2.25 gallons per minute.

A reinspection of the bore hole was carried out a year after it was drilled. The investigation consisted of a CCTV survey in side the hole during each of the following stages of penstock operation:

. while the unit was operating at full capacity,
. when the penstock was primed with a static water head, and
. while the unit was shutdown and dewatered.

Gaps in the horizontal and vertical planes in three of the steps of the construction joint intercepted by the bore hole were measured at all the three stages. No visible change was noticed in the width of the joint during these stages. There was a noticeable decrease in the width of the joint from the previous year but the water flow had increased to 5.1 gallons per minute from the previous year's flow of 2.25 gallons per minute. This was probably due to the washing away of the debris in the joint during the year. The water leakage stopped almost immediately after the unit was dewatered.

Based on the observations it was concluded that the deformation is probably permanent and does not depend on the load and can not be closed by prestressing etc. It was confirmed that the leak originates within the transition section and does not come through the joints in the main dam.

4.4 CONCLUSIONS:

The review of Des Joachims G.S. penstock behaviour indicates that a large number of factors combined together are responsible for the joint and concrete deterioration and the resulting leakage.

Lack of steel liner and water stops at joints were obvious reasons. In addition, the stepped construction joint with a number of corners-without proper reinforcements- made it easy for the cracks to initiate. Less than adequate reinforcements on the side walls due to the assumed higher allowable rebar stress and poor lap splicing probably were the reasons for the opening up of the joints on the walls. Also the loads which were not considered in the original design namely, temperature effect, pore pressure at joints, seismic forces, effect of dam movements and creep and shrinkage effects appear to have contributed to the problem.

Some poor construction features suspected were: early shrinkage, poor bonding at joints and inadequate joint details.

5.0 SUMMARY FOR ALL STATIONS:

Analytical review and field inspections were carried out in detail at all other stations as well. However, no other bore hole investigation has been done.

The review indicates that although some differences exist between stations all stations which have the same features as Des Joachims G.S. have similar problems. Stations which have the step joint construction have cracks and leakage. Penstock transitions with smooth wall joints have less cracking.

6.0 RESTORATION/REPAIR OPTIONS:

A preliminary study of the repair options were carried out considering the required structural integrity, leak tightness and construction feasibility. The construction cost was not specifically considered at this stage. The options studied are:

Remove the top concrete (see shaded area of Figure 1(b)) and replace using full steel liner: This can be designed to satisfy all the engineering requirements. Obviously the front end cost will be high. Installing steel liner without removing concrete top does not appear to be feasible because of the difficult access and confined space. The main concern in this option is the proper placement and attachment of the liner to existing concrete.

Anchoring the top half by post tensioning and Crack grouting: In this approach, the top section (the shaded area of concrete in Figure 1(b)) will be anchored to the bottom half and/or the bed rock by post tensioned anchors and all the cracks and joints will be grouted. This approach will satisfy the strength requirements. However, the ability to achieve durable leak tightness is questionable.

Provide Flexible Liner: A flexible liner which can be installed at site and is durable would be an ideal solution to prevent leakage. The requirements of a flexible liner are that it should be capable of being easily applied, should adhere to dry/wet surface, resist abrasion and flow induced forces, flexible to accomodate crack and joint movement and durable. Some possible approaches for a flexible liner are:

Fiberglass Liners: In this method a flexible fiberglass lining of about 1/2 inch thick would be applied at site to the interior of the concrete section. This method has been successfully used to repair steel penstocks (Graham and Kahl, 1989). Use of such liners for concrete and the durability of fiberglass liners need more investigation.

Prefabricated Membrane Liners: In recent years use of chemically cured, elastomeric liner and resin impregnated insitu forms have been successfully used to provide leak tightness. While such liners have been used in sewer pipes, tanks etc. we are not aware of their use in hydraulic structures requiring resistance to high velocity flow, abrasion and leaktightness.

Liner Coatings: A number of elstomeric coatings are currently in the market which claim excellent properties suitable for penstock applications.

Polyurethane Resin Curtain: An internal liner curtain can be created by injecting polyurethane resin to the desired depth (Smoak, 1991).

Based on the severity of the problem and deficiencies at each station, four different types of repair options were suggested as given in Table 1. Cost of the repair option were not explicitly considered in making these recommendations at this stage. However, it was to be performed prior to selecting the options suggested. Detailed investigation of the properties of flexible liners was also recommended.

Table 1: Suggested Repair Methods

Stations	Repair Method
Des Joachims Pine Portage Red Rock	Remove concrete and replace using steel liner*
Stewertville 1,2 Mountain Chute Rayner	Post Tension + grout + flexible liner
Barrett Chute 1,2	Post Tension + grout
Aubrey Falls Wells Otter Rapids	Grout + surface repair
Stewertville 4,5 Barrett Chute 3,4	none required

* flexible liner shall also be investigated

6.0 SUMMARY:

Concrete cracking, joint deterioration and leakage in a number of unlined concrete penstock transition sections were investigated to determine the causes and suggest remedial measures. A number of deficiencies in the original design and construction features were identified. Possible repair/rehabilitation methods were proposed depending on the severity of the problem.

REFERENCES:

1. Graham Douglas, Kahl Tom, "Pittsford Penstock Rehabilitation", Hydro Review, April 1989, pp. 36-41.

2. Smoak Glenn W., "Crack Repairs to Upper Stillwater Dam", ACI, Concrete International, Feb. 1991, pp. 33-36.

FORT PECK - POWER PLANT NO.1 PENSTOCK REPLACEMENT

Ronald W. Bockerman, PE[1] and Donald F. Miller[2]

ABSTRACT

The U S Army Corps of Engineers Fort Peck hydro facility consists of two adjacent powerhouses, with a combined capacity of 185 MW, located on the Missouri River in the state of Montana, USA. Power Plant No.1 was placed in service in 1943 followed by Power Plant No.2 in 1961. The purpose of this rehabilitation work was to replace structurally substandard penstocks (with no articulation joints); to eliminate the potential of overtopping the 60 meter high surge tanks; to replace the 50 year old isolation valves; to increase the plant capacity by improving flow efficiencies; and to renew the failing corrosion protection coatings on the turbines, surge tanks, and supply tunnel steel liner.

INTRODUCTION

The Fort Peck Project is located on the Missouri River in northeast Montana, 35 kilometers southeast of Glasgow. The dam is the largest hydraulic fill embankment in the world and the lake is the fifth largest man-made reservoir in the United States. Construction started in 1933 and consists of the dam, two power plants, spillway, two flood control tunnels, and ancillary service buildings. The power produced by this multi-purpose project is distributed by the Western Area Power Administration (WAPA).

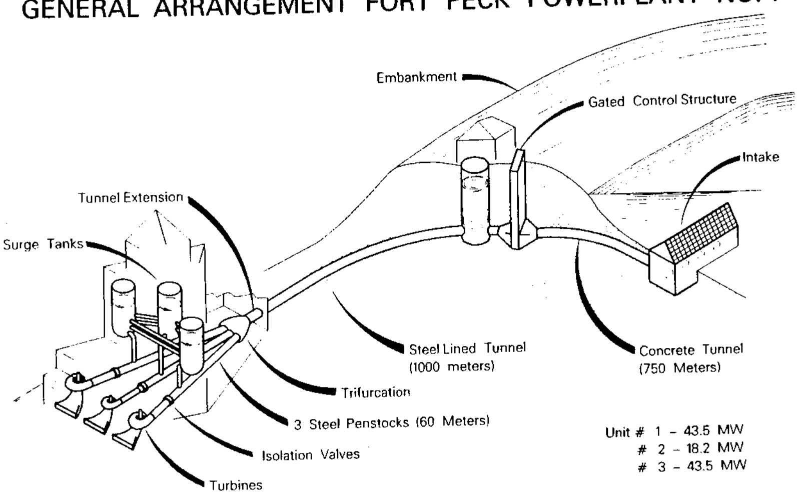
Power Plant No.1 consists of a screened, low-level intake supplying three Francis turbines via 750 meters of reinforced concrete tunnel, gated control structure, 1,000 meters of riveted steel pressure tunnel, a trifurcation section, and three penstocks. The power tunnel and

---

[1]Project Manager, Hydro Power Branch, Operations Division
[2]Technical Manager, Design Branch, Engineering Division
U S Army Corps of Engineers, Omaha District
215 North 17th Street, Omaha, NE 68102-4978

penstocks are 7.5 meters and 4.25 meters in diameter, respectively. Each penstock has an isolation valve positioned between the turbine and surge tank riser to permit individual units to be unwatered for maintenance.

GENERAL ARRANGEMENT FORT PECK POWERPLANT NO. 1



The present powerhouse was constructed in stages, beginning with a single 35 MW unit in the early 1940's, then adding a trifurcation section, surge tanks, a second 15 MW unit, and part of Penstock No.3 in 1943. The third 35 MW unit was added in 1951. Studies completed in 1975 led to rewinding the generator stators and uprating the units to 43.5, 18.2, & 43.5 MW, respectively.

The tunnel and surge tank were hydraulically model tested in 1940-41 to verify surge tank sizing and to determine water hammer and hydraulic friction losses. In 1952, testing was undertaken to determine the efficiency of Units 2 and 3. This testing indicated that the actual discharge of the units was significantly higher than the turbine model tests had predicted.

Following the uprating, 1978 computer modeling and load rejection testing indicated that the surge tanks could be overtopped. To prevent such a potential catastrophic occurrence, discharge restrictions and loading rate limitations were imposed on the plant. Restricting the risers would prevent overtopping, however, this would increase water hammer pressure in the penstocks.

Prior to modifying the riser orifices, a thorough inspection and preliminary analysis of the penstocks were conducted to determine the adequacy of the system to sustain increased pressure. The results proved to be less

than desirable. First, extensive welding had been done to the butt straps and rivet heads at sometime in the plant's early history, presumably to stop leaks. No specifications or records could be found to document that this welding was done using controlled procedures. Second, the penstock system was constructed without articulation joints to allow for differential settlement of the plant or thermal expansion and contraction of the penstocks. Third, the penstock system was fabricated from a mixture of steel specifications furnished under three separate contracts during World War II when there were extensive problems with steel allocations. In fact, construction documents stated: "that when materials included in the original design could not be obtained, a hurried redesign was necessary to eliminate critical materials and provide for the use of substitute materials".

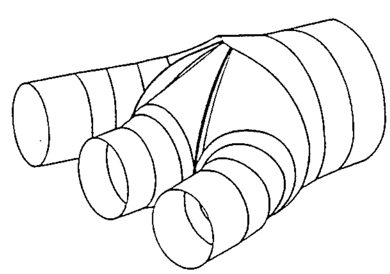
Inspection of the riveted steel supply tunnel showed the 50 year old existing coal tar coating missing in places, revealing the start of pitting corrosion. The abutment where the tunnel was constructed consists of a shale formation with thinly bedded bentonite seams. It was determined that seepage from the tunnel could cause instability in the shale formation, fail the tunnel, and lead to an embankment failure. At this time it was decided to recoat the tunnel with a vinyl paint system.

In 1979, the Shawinigan Engineering Corporation, San Francisco, California, was asked to evaluate the condition of the penstock system and to recommend a permanent solution for the penstock and surge tank problems. In 1981 they submitted a final report with the following result. Due to the lack of expansion/contraction joints, the rivets in the circumferential joints were under designed for shear during normal loading. The lowest calculated Factor of Safety (F.S.) was 1.8, substantially below the 4.0 required by Corps of Engineers design criteria and other codes for riveted steel construction. Shawinigan offered several potential solutions but, in conclusion, the report stated that the only option reasonably certain to insure full head operation of the plant was to further restrict the surge tank risers and replace the penstock system.

In March 1990, the District analyzed the effect of differential settlements between the powerhouse, trifurcation, and tunnel extension monoliths. A geotechnical investigation of the shale foundation indicated that a settlement of 10mm between monoliths occurred during the 50-year life of the project. Although this settlement was small, it subjected the penstocks to 106 MPa (15,400 psi) of additional bending stress and created additional rivet shear not considered in Shawinigan's report. Recalculation of the F.S. due to increased rivet shear revealed that the F.S. was as low as 1.2 even with the imposed operating restrictions.

PENSTOCK REPLACEMENT

On April 13, 1990, with the above indications that a catastrophic failure of the penstock system could be imminent, the plant was taken out of service. A year prior to that, R.W.Beck and Associates, Seattle, Washington, had been hired to prepare plans and specifications to remove the existing penstock system and to replace it with new welded steel construction. The proposed design fitted all three penstocks with expansion/contraction joints and ring girder supports with rocker bearings to allow for differential settlements. The surge tank risers were designed with adjustable orifices that could be easily modified following proto-type testing of the new system. R.W.Beck retained Sultzer-Escher Wyss, Zurich, Switzerland, as their consultant for the trifurcation design. The Sultzer-Escher Wyss patented trifurcation design utilizes internal stiffener rings in the crotch, rather than the conventional external rings. The Corps of Engineers, Waterways Experiment Station (WES), Vicksburg, Mississippi, was tasked to hydraulically model test the new design.



TRIFURCATION

Also prior to shutdown, it had been decided to replace the existing butterfly type, isolation valves. This decision was based on the fact that, after 50 years of service, the valves were reaching the end of their design life and replacement of the existing valves in later years would require costly removal of portions of the new penstock system. To facilitate future replacement of the smaller Unit No.2 with a larger unit, all three new penstocks and isolation valves were sized the same.

With the plant off line came the requirement to obtain immediate funding and to accelerate the schedule to do the rehabilitation in the shortest possible time. Fortunately, laws already in effect permitted WAPA to directly reimburse the Omaha District as the work progressed, thus avoiding the normal lengthy budgeting process. Within days, a compressed schedule was developed that involved continued design work by the Omaha District, R.W. Beck, and WES, that led to the award and completion of one supply contract and two construction contracts. Approximately one year was trimmed from the original schedule with the goal of plant back on line by October 1, 1992, with the ultimate requirement of December 1, 1992 to prevent the purchase of $1,200,000 of power capacity by WAPA.

To make the schedule work, two major milestones had to be met. The tunnel painting contract had to be awarded in time to mobilize and remove the existing coal tar enamel coating in the winter of 90/91 and to complete the painting before the arrival of the following winter weather. This would also minimize the time that the painting contractor and the penstock contractor would be occupying the same space. Second, because of the long lead time required, the valve fabrication had to begin as soon as possible to meet a delivery time critical to the penstock replacement contract. Because of the location of the valves, the installation of the new penstock components had to be done in a certain order. Late delivery of the valves would have meant virtual shutdown of the penstock work.

Our procurement task allowed us to use either the Invitation For Bid (IFB) type contract, where the low bidder is chosen, or the more time consuming Request For Proposals (RFP) method of obtaining contracts, where the proposals, contractors, and bids are evaluated and compared with the winner not necessarily the low bidder. It was decided to invest a little more time up front with the RFP, choose the best Contractors, and reduce the risk of delays during construction. We are convinced that the decision was correct. Of the three contracts, one was awarded to the low bidder and two were awarded to second low bidders.

The tunnel painting contract was awarded September 24, 1990 to a joint venture of Venture Construction and Interstate Coatings, Seattle, Washington, for $2,702,309, with a contract completion date of December 31, 1991.

The Omaha District has had numerous contracts over the last ten years involving the removal of old coal tar enamel. After the initial experiences, we have advised succeeding contractors that the most expedient method of removal is simply to strike the surface about every three or four inches with ball peen hammers. We have yet to convince a Contractor of that until they have made an attempt to do it by some sort of media blasting. As in all previous cases, there was a run on hammers at the local hardware stores and a lot of sore arms.

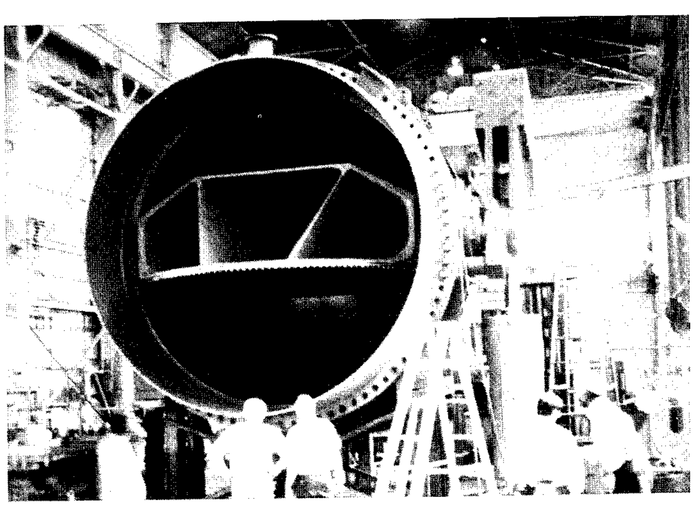
After the coal tar was removed, one innovation that the contractor tried that we hoped would prove successful, was the use of robotic type machinery to do the blasting and painting. Unfortunately, the riveted plate construction proved to be too much for the system that was tried and conventional methods were substituted. However, the experiment indicated that the machinery, which consisted of a system of rotating heads mounted on a wheeled vehicle that was to be pulled down the tunnel, might adapt very well to welded steel construction. Such an arrangement may not only reduce time, cost, and provide more even surface preparation and paint application, but

also would reduce human exposure to the hazardous atmosphere created by blasting and painting.

The tunnel surface was blasted to a "near white" condition using a recoverable steel grit and covered with a four coat vinyl system. In addition to the tunnel, this contract included blasting and vinyl painting the exterior surfaces of the three surge tanks. This work was difficult because, not only are the tanks riveted plate construction and supported with external structural steel members, but the existing coating had a lead based primer requiring special handling and disposal of the contaminated blasting media. The three turbine runners, discharge rings, and scroll cases were also recoated. This contract was completed on time and required only two modifications resulting in a net credit to the Government of $9,811.

Two weeks after the tunnel contract was awarded, a supply contract to design, fabricate, and deliver the three, 4.25 meter diameter, butterfly valves was awarded to Kvaerner Hydro Power, San Fransico, California. The design of the horizontal shaft, flow through disc, hydraulic cylinder operated valve was done by Kvaerner staff in Doncaster, England. Each disc incorporated two large, steel castings which were produced by the Maynard Steel Casting Company, Milwaukee, Wisconsin.

The castings were shipped by truck to the Hardie-Tynes Manufacturing Company, Birmingham, Alabama, where the valves and valve extensions were fabricated, assembled, and factory tested. The tests included welding temporary, dish shaped test heads to the ends of the valve extensions and



hydrostatic testing the assembly at 1550 KPa (225 psi). Following the testing, the flange connected valve extensions were removed to reduce the shipping weight but the main valve assembly was shipped intact and as tested to the construction site. The hydraulic controls were provided by Hydra-Power Systems, Inc., Portland, Oregon.

The valves were delivered individually, one month apart, all ahead of schedule. At the time of this printing, this $3,920,000 contract has not been completely finalized but, again, there were few modifications (three) and final cost to the Government will be no more than the original contract price.

The third and final contract was awarded to CBI Services, Inc., Fremont, California, on January 7, 1991, for $9,413,000. However, before the Notice To Proceed (NTP) could be issued, one of the unsuccessful proposers protested the award. The protest was eventually denied but the NTP was delayed two months thus delaying the plant back on line date to the critical December 1, 1992 deadline. Therefore, all contingency time was exhausted before the Contractor was even allowed to mobilize and the exact contract schedule had to be maintained.

CBI subcontracted the demolition work to Moltz Constructors Inc., Cody, Wyoming. While painting was being completed in the tunnel, the existing tunnel extension, concrete embedded trifurcation section, penstocks, and butterfly valves were cut into sections and removed as scrap. The penstock system is located below ground level and, to facilitate removal of the existing and installation of the new, an access opening was cut in the concrete deck that covers the penstock gallery.

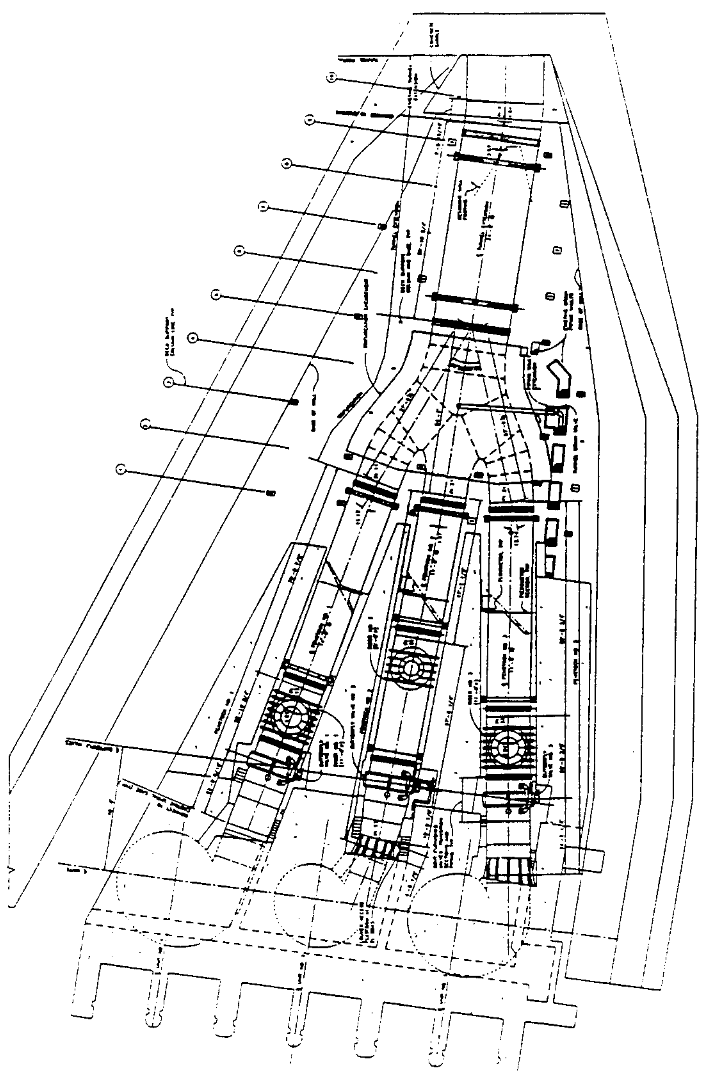
CBI subcontracted with Eaton Metal Products Company to fabricate sections of the penstock and tunnel extension in plants located in Denver, Colorado and Salt Lake City, Utah. The penstock was fabricated in sections that spanned between the new expansion joints as much as practical to limit field welding. Sections 4.25 meters in diameter and as long as 12.6 meters, complete with ring girders, were shipped by truck to the site.

The 186 metric ton trifurcation was fabricated at CBI's plant in Kankankee, Illinois. The trifurcation shell plate thickness varies from 22mm to 48mm and the internal ring stiffener is 102mm thick. Due to the enormity of the trifurcation, it was fabricated in sub-assemblies suitable for shipment by truck. To avoid fit-up problems in the field, it was completely shop assembled and piece marked prior to shipping.

CBI simultaneously started installing the new penstock sections at the entrance to the scroll cases and the new tunnel extension at the downstream end of the power tunnel. Since both the scroll cases and tunnel are embedded in concrete, the riveted steel had been cut 1/2 meter past the concrete. At these locations, although not required by welding codes, the connections of the new steel to the existing were both pre- and post-weld heat treated due to the uncertainty of the chemical characteristics of the riveted steel. Work progressed in both directions towards the trifurcation where the new system would be tied together.

Due to space limitations in the penstock gallery, penstock cans were welded into complete spools at ground level where practical. These sections were then lowered into the penstock gallery, placed on temporary track, and moved into final position. The trifurcation was field assembled into four major sub-assembles which were then lowered into the penstock gallery for final fit-up. Due to the thickness of the plate material in the trifurcation, post-weld heat treatment was specified after final assembly. CBI constructed a ceramic fiber lined, steel enclosure around the trifurcation to act as a furnace. The same enclosure had previously been used at ground level to provide a shelter for the sub-assembly welding.

Installation of the valves required that they be lowered 20 meters to the penstock gallery floor and moved laterally downstream into existing concrete corridors. Using the track noted above, CBI assembled wheeled carriers that not only moved the 75 metric ton valves through these narrow passages but also rotated them at one point along the way to clear existing obstructions. Once in place the valves were permanently supported from the floor and the cylinder, counter weight, and other accessories were added.



NEW CONSTRUCTION - PLAN VIEW

Non-destructive testing requirements for the penstock system required 100 % radiographic inspection of all welds. Due to the high stresses on the internal stiffener rings of the trifurcation, magnetic particle, dye penetrant, and ultrasonic testing was done in addition to radiographic inspection. The penstock system was also hydrostatic tested at 1550 KPa, 1.5 times design pressure. Following these tests, the trifurcation was embedded in mass concrete for stability and dampening requirements.

The power plant went back on line December 1, 1992 as scheduled with only insignificant items to be completed in the spring. The total cost of the modifications to CBI's contract was $103,649, only 1.1 % of original contract price, and there were no time extensions.

CONCLUDING REMARKS

The total cost of the project was $18,500,000 and it took approximately four years to complete. River flow and limited power production was maintained through Power Plant No.2 while Plant No.1 was out of service for 2 1/2 years. Accusonic Inc, acoustic flow meters were installed in the new penstock sections. The work was accomplished with no adverse effect on the environment. Much of the off site Quality Assurance was performed by DCASMA. The project was managed under the Corps of Engineers Project Management program and, although the schedule was very compressed, the work was always on schedule and within budget.

At the time this paper was written, the proto type tests were being scheduled for the spring of 1993 and the results will be included in the August presentation at the WATERPOWER '93 conference.

US Army Corps
of Engineers

ELIMINATION OF SURGE TANKS AT SALUDA HYDROELECTRIC PLANT

Patrick Ward[1], Timothy Lynch[2], Fred Harty[3] - SWEC
Kristina Massey[4], Hal Riddle[5] - SCE&G

Abstract

This paper discusses the elimination of two 1930s-vintage surge tanks at a hydroelectric plant in South Carolina that is interconnected hydraulically with the circulating water system of a steam electric generating station. The paper discusses the planning, design, construction, and testing phases of the modifications. The planning phase included the evaluation of various alternatives for eliminating the surge tanks while still accommodating the circulating water pipe take offs. During the design phase, the required engineering analyses were performed to facilitate designing the chosen alternative. In addition, in this phase challenges were faced in interfacing new material with existing material. Construction and testing presented their own challenges.

Background

The Saluda Hydroelectric Plant began operation in 1930 with four 32 MW Francis-type turbine generator units. Two of these units (Units 1 and 3) had surge tanks to permit rapid load change. The other two units (Units 2 and 4) did not have surge tanks, but had tee stubs to which surge

---

[1] P. Ward - Senior Mechanical Engineer - Stone & Webster Engineering Corporation, 245 Summer St., Boston, MA 02210.
[2] T. Lynch - Principal Structural Engineer - Stone & Webster, Boston, MA.
[3] F. Harty - Senior Principal Civil Engineer - Stone & Webster, Boston, MA.
[4] K. Massey - Senior Engineer - South Carolina Electric & Gas Co.
[5] H. Riddle - Engineer - South Carolina Electric & Gas Co.

tanks could be connected later. The two Johnson-type surge tanks were 38 ft (11.58 m) in diameter and extended 190 ft (57.9 m) above the ground surface. Looking like huge inverted wine bottles, these massive structures dominated the skyline near the plant (see Photos 1 & 2).

In 1958, the McMeekin Coal-Fired Station was constructed adjacent to the Saluda Hydroelectric Plant and its circulating water system was interconnected hydraulically with the Saluda Hydroelectric Plant surge tanks. The McMeekin circulating water system drew cool water from a 90-inch (2,286 mm) diameter pipe leading to the Saluda Unit 1 surge tank riser and returned heated water to the Saluda Unit 3 surge tank riser. Valves also permitted the McMeekin system to draw from the Saluda Unit 3 surge tank riser, then discharge the return flow through a Howell Bunger valve to the Saluda tailrace. The latter arrangement was used when required to perform maintenance at the Saluda Plant.

Maintenance costs of the surge tanks had become significant over the years. South Carolina Electric & Gas Company (SCE&G), the owner of the Saluda plant, determined that slower wicket gate timing would be acceptable for Units 1 and 3 as a tradeoff for reduced maintenance costs without the surge tanks. The next step was to determine whether acceptable pressures could be maintained in the combined hydraulic system without the surge tanks. All four turbine-generator units have identical construction. Therefore, Units 1 and 3 could be expected to resist the same over-speeds as Units 2 and 4, which already had the slower wicket gate timing.

Stone & Webster Engineering Corporation (Stone & Webster) was retained to perform the hydraulic transient analyses, to develop the conceptual and detailed designs of the riser modifications, and to confirm that the modified system operates as predicted. SCE&G worked closely with Stone & Webster during project development and performed all construction management and operation activities.

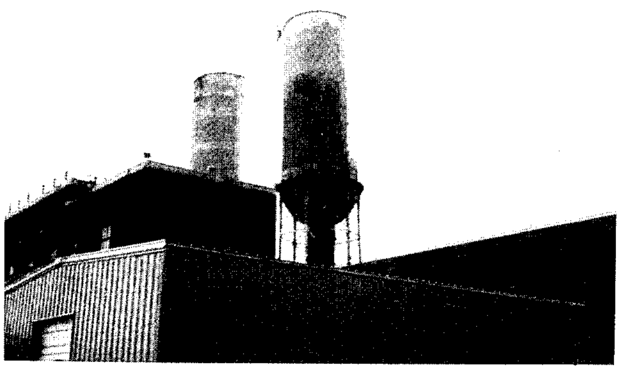


Photo 1. The two surge tanks prior to removal.

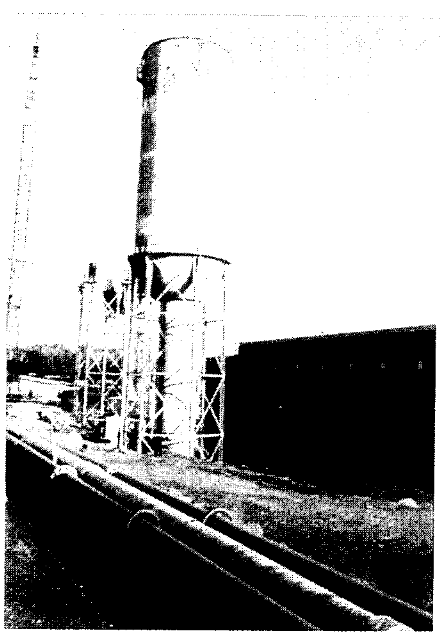


Photo 2. View after removal of one surge tank.

Planning

Hydraulic transient evaluations were performed to confirm the acceptability of the pressures in the combined systems anticipated after surge tank removal. Next, the alternatives for eliminating the surge tanks were evaluated conceptually to determine which one would be the most effective.

Two alternatives for capping and rehabilitating the surge tank risers while retaining the circulating water supply and return connections for the adjacent McMeekin station were compared. Alternative 1 (see Figure 1) features the installation of a new, smaller diameter riser, approximately 7 ft-6 in. (2.29 m) in diameter. It would be oriented vertically and encased in new concrete that would be attached to the existing concrete encasement of the old riser. The new riser would be connected to the existing surge tank riser about 3 ft (0.91 m) above the top of the penstock. The space between the new riser and the existing 14 ft-0 in. (4.27 m) diameter riser would be filled with new concrete anchored through the existing steel riser to the existing concrete. The new riser would be attached to the existing circulating water pipe with a mitered elbow at approximately the same location as the penetration of the existing surge tank riser.

Alternative 2 (see Figure 2) features the installation of a new riser stub approximately 13 ft-6 in. (4.11 m) in diameter and only slightly smaller than the existing riser. It would be capped above the circulating water pipe penetration and extend below the top of the concrete to elevation 178 ft (54.25 m), the same level as Alternative 1. The new riser stub would be anchored to existing concrete and grout would be placed between the new riser and the existing riser.

The alternatives were evaluated based on an internal design pressure of 135 psi (0.93 N/mm^2). This pressure was selected to be compatible with the design pressure of the interconnected circulating water system. The governing design pressure in the Saluda system was 102 psi (0.70 N/mm^2) at the surge tank tee connection. Maximum pressure was expected to occur during load rejection and shutdown of the Saluda Hydroelectric Plant units with a maximum headpond elevation of 360 ft (109.73 m).

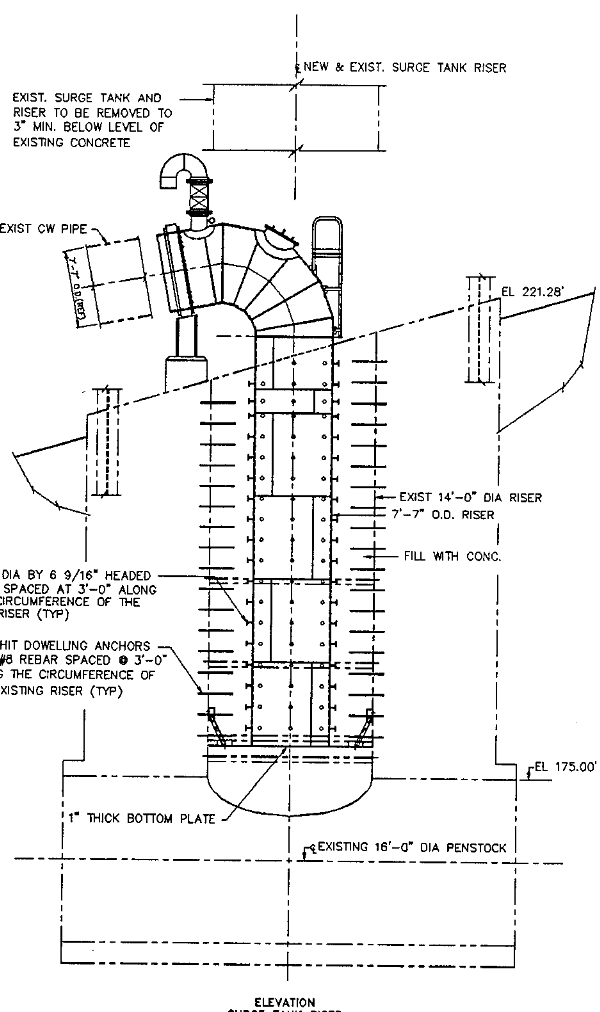


Figure 1. Alternative 1 - Installation of a new, smaller diameter riser.

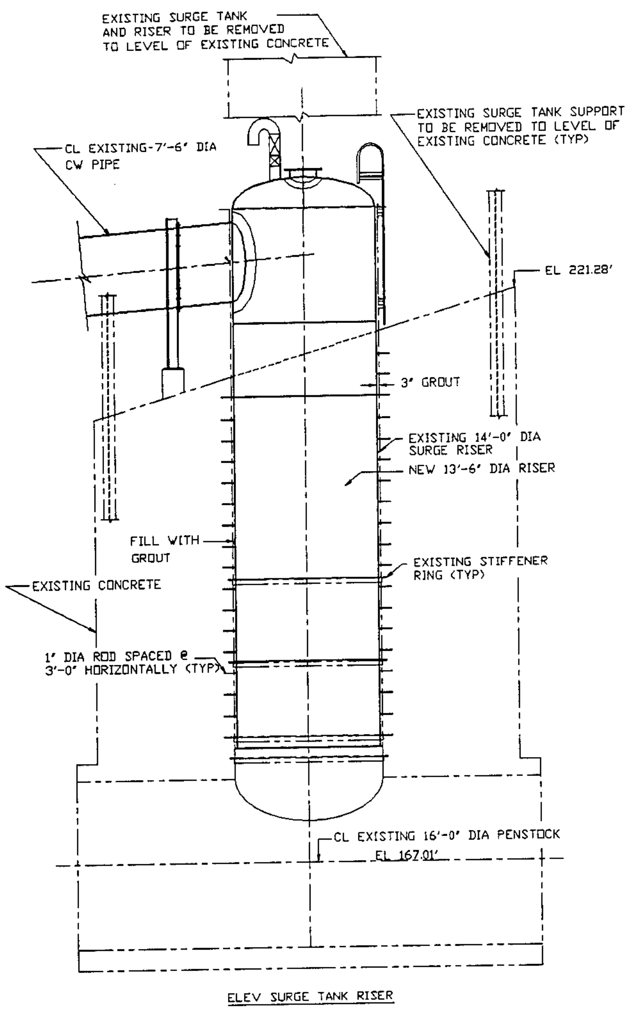


Figure 2. Alternative 2 - Installation of a new riser stub.

The designs of both alternatives considered structural loading due to the upward hydraulic thrust from the new closure pieces and the horizontal thrust from the circulating water pipe. The upward thrust was to be resisted by engaging the weight of the concrete block which encases the riser. The riser elbow was also designed to withstand a vacuum of 2 psi (0.0007 N/mm^2). A ball check vent valve was provided at the top of the new riser elbow or cap to open automatically to admit air, and close automatically when the riser was full of water. A manual gate valve between the riser and the ball check vent valve would facilitate maintenance. The gate valve would remain open during normal operation, but would be closed only for maintenance.

Based on the results of the study, Alternative 1 was chosen. The comparative cost estimates showed that the capital cost for the 7 ft-6 in. (2.29 m) riser was significantly lower than the capital cost for the 13 ft-6 in. (4.11 m) riser called for in Alternative 2. Also, the 7 ft-6 in. (2.29 m) riser could be constructed more quickly than the 13 ft-6 in. (4.11 m) riser because the 13 ft-6 in. (4.11 m) riser would require more field fabrication. The estimated outage duration and associated cost of downtime was essentially the same for both alternatives.

Design of the Chosen Alternative

The existing circulating water pipe had been connected to the surge tank through a 7 ft-6 in. (2.29 m) stub and a slip connection with a packing gland-type seal. This joint allowed for axial movement due to thermal and pressure changes in the circulating water pipe. Vertical thrust was not a factor because the surge tank was open to the atmosphere. The design of the new 7 ft-6 in. (2.29 m) riser needed to address the thrust from the existing circulating water pipe connection, plus vertical thrust on the closed elbow and the restraint of embedment.

The embedded section of the new riser has headed anchors attached to the outer surface which are embedded in the new concrete. These anchors transfer the upward hydraulic thrust from the riser to the concrete. Additional resistance due to bond was deemed insignificant. The upward thrust on the new concrete is transferred to the existing concrete using deformed anchor bars installed through the existing riser and into the existing concrete.

The top of the 7 ft-6 in. (2.29 m) riser is mitered, as required by piping codes applicable to large bore fabricated pipe, to provide uniform stresses in the riser. The mitered elbow also contained a maintenance manhole with a reinforcing plate.

The riser was modeled using the finite element analysis (FEA) method. FEA was chosen due to the complex system configuration including the manhole, the effect of concrete embedment, and the effect of the slip joint at the connection with the circulating water pipe. The riser was modelled from the top of the concrete to the connection with the circulating water pipe. The interface with the concrete was modeled as a fixed end condition to develop a conservative estimate of the stress in the riser. The slip joint was modeled by restraining the radial growth and lateral (upward) displacement while allowing axial displacement. The restraint from the slip joint, combined with the weight of the water and the pipe, provides a secure connection.

In the final design, stresses were limited to the lesser of 67 percent of the yield stress and 33 percent of the ultimate tensile stress. Hoop, longitudinal, and secondary bending stresses were combined in accordance with the Hencky-Mises theory of failure using the following equation: $Se^2 = Sx^2 - SxSy + Sy^2$, where Se equals the equivalent stress and Sx and Sy are the principal stresses. The equivalent stress obtained from the Hencky-Mises theory was not permitted to exceed the allowable stress. Supports were designed in accordance with the AISC Manual of Steel Construction, 9th edition. Concrete and concrete anchors were designed in accordance with the Building Code for Reinforced Concrete, ACI 318.

Construction

A single contractor fabricated and installed the new 7 ft-6 in. (2.29 m) riser (see Photos 3 & 4). This provided a single point of responsibility for dimensional checks, fabrication tolerances, delivery, construction coordination, and interface control. The Contractor dismantled existing parts required for installation of the new riser and provided assistance to SCE&G when the affected unit was returned to service.

A key challenge faced during construction was how to complete the modifications while limiting outage. To meet this challenge, the units were taken out of service one at a time, and the first unit was returned to service before the outage was initiated on the second unit.

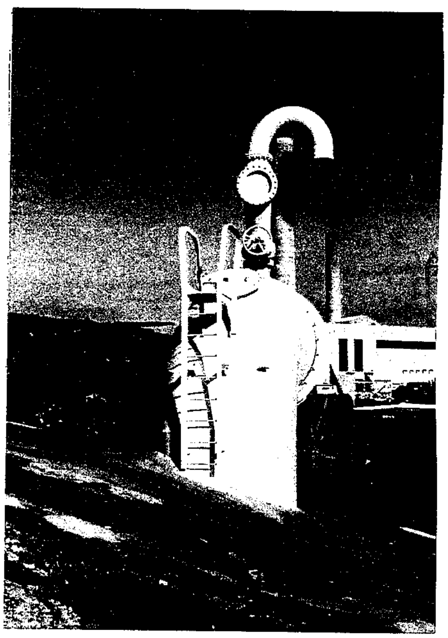


Photo 3. Fabrication and installation of new riser.

Photo 4. Fabrication and installation of new riser.

Tests

Prior to confirming results, load rejection tests were performed before surge tank removal to provide baseline data. Confirmatory load rejection tests were performed following the removal of the first surge tank and again following the removal of the second surge tank. Wicket gate timing, speed rise, and pressures were measured at designated points in the water conduit system. The data were analyzed on site between tests and stored on magnetic tape for possible additional analysis, if required. The test results confirmed the acceptability of the modifications.

MODERNIZATION OF PELTON TURBINE PRESSURE REGULATORS

Louis G. Silva[1] and Difa Shveyd[2]

ABSTRACT

Hydroelectric facilities are aging assets within the infrastructure of many electric utilities. As a consequence, many of those facilities are in need of modernization.

This paper presents four examples of the replacement of Pelton type turbine pressure regulator valves with jet deflector systems. The jet deflectors reduced the pressure rise within the penstock, reduced the speed rise in the turbine-generator and provided more positive control of the turbine operating system.

The interdisciplinary nature of this type of rehabilitation is described. The decision to proceed with system improvements includes the consideration of electrical, mechanical, civil and environmental conditions. Given the age of most older hydroelectric systems, a comprehensive evaluation of major system components, their reliability and estimated remaining life must be completed before improvements are made. After it has been determined which of the existing system components, including the penstock are structurally and mechanically suitable for modernization, the process of designing and retro-fitting can proceed.

Several considerations are reviewed, including adequacy of civil foundations and their effects on proposed mechanical modifications. In addition, the effects of environmental concerns are reviewed.

The paper presents case studies of rehabilitation programs which provided improved system reliability and safety, in a cost effective and environmentally sound manner.

[1] Civil Engineer, Pacific Gas & Electric Company, San Francisco, California
[2] Mechanical Engineer, Pacific Gas & Electric Company, San Francisco, California

HISTORICAL DATA

Pelton turbines are of the impulse class and develop their power by first converting the water's energy into a jet issuing from a power nozzle. The jet impacts against a series of bowl-shaped buckets on the runner periphery to produce driving torque to rotate the turbine-generator.

The sudden loss of electrical load on a hydro turbine-generator unit can produce a rapid and potentially harmful speed rise for the generator. On Pelton turbines this speed rise is usually limited to a safe value by quickly diverting the water jet away from the runner on load rejection. While rapid closure of the jet controlling power needle would limit speed rise, it would have the unacceptable effect of producing a high water hammer pressure rise in the penstock.

To address this problem, some early Pelton turbines were equipped with either auxiliary nozzles or angle valve pressure regulators designed to rapidly open as the power nozzle closed upon load rejection. This limited speed rise while avoiding a rapid change in penstock flow with its attendant pressure rise. Another method employed a swivel joint on the power nozzle which allowed the jet to be diverted away from the runner by physically deflecting the power nozzle upon load rejection. Maintenance of the swivel joint was difficult and control of the diverted jet within the tailrace were problems in this scheme.

The modern solution using jet deflectors dates back to about 1910 and effectively made the drop nozzles obsolete. Pelton turbines with auxiliary nozzles continued to be supplied until about 1920. Since 1930 most Pelton turbines have been furnished with jet deflectors for speed and pressure rise control.

PELTON TURBINES IN THE PG&E SYSTEM

PG&E's hydro facilities located mostly in the Sierra Nevada Mountain Range include some thirty-six Pelton turbines, eighteen of which were supplied without jet deflectors. While these older units with auxiliary or drop nozzles, and/or pressure regulators may have originally provided acceptable speed and pressure rise control; some of them have experienced excessive wear and have required increased maintenance. More importantly, wear in the auxiliary nozzles or pressure regulators resulted in higher pressure rises than considered acceptable. Accordingly PG&E undertook a program to address this problem.

SOLUTION ALTERNATIVES

Penstock pressure charts, maintenance records, and available data on load rejection tests were evaluated relative to penstock pressure rises and turbine-generator speed rises for a number of powerhouses with Pelton turbines with auxiliary nozzles or pressure regulators. Penstock extreme pressure rise data was examined in light of the

probable cause and the likelihood that the speed and pressure transients could be reduced by installation of deflectors or other mechanical equipment.

CARIBOU NO. 1: The original Caribou 1 penstock was installed by Great Western Power Company in 1921 and 1923. The original pressure regulator valves were designed to limit the maximum pressure rise during normal load rejection for the penstock as designed when the turbine was ordered. During construction the location of the surge tanks was moved upstream increasing the penstock length. Load rejection tests showed pressure rises in excess of 27% above static. The corresponding maximum turbine-generator speed rise was 56.9% above synchronous speed. Normally in systems with pressure regulator valves the speed rise is limited to 30% above synchronous. The penstock pressure rise is limited to 10% above static. As a result of these tests the units were restricted to 92% of their capacity, with approximately equal loading on each wheel, was placed on the units. This limited the speed rise to not more than 40% and the pressure rise to not more that 10% following full load rejection. The operation of these units below 70% of capacity was restricted also due to wear in governor and pressure regulator valve. High pressure rises occurred both during load rejections and start-up of unit caused by mismatch between power needle and pressure regulator valve at small needle opening.

WISHON: Studies of the Wishon penstock data identified two previous extreme pressure events. In 1986, the penstock recording chart indicated a pressure down surge from an initial pressure of 600 psi to 200 psi with a subsequent upsurge to over 1,000 psi (recorded at the turbine nozzle). A similar event occurred in 1943.

TULE: At Tule Powerhouse, the penstock experienced a high pressure rise initiated by debris clogging the control water lines. The malfunction resulted in rapid full flow shutoff by the turbine and opening of the pressure regulator valve. The penstock pressure chart recorded pressure rises in excess of 1,000 psi representing a pressure rise of 50% above static.

BALCH NO. 1: The Balch 1 penstock experienced an extreme pressure rise resulting from governor oscillations near zero gate. The power needle oscillated near speed-no-load opening at the natural period of the penstock system (2.2 seconds). The penstock pressure chart recorded pressures in excess of 1,300 psi, approaching 30% above static. Adjustments were made to the governors to prevent oscillations, and to limit pressure rises to 10% above static.

Following this review, two alternatives were considered to correct these problems:

Alt. 1 - Refurbish existing pressure regulator system.
Alt. 2 - Install deflectors and limit needle closure rate.

Based on safety considerations and overall reliability of the deflector systems, Alternative 2 was selected.

DEFLECTOR DESIGN

MECHANICAL: Since powerhouses that are candidates for upgrades or reconditioning are generally quite old, few original or accurate construction drawings, or drawings of mechanical equipment are available.

Based on experience with jet deflector systems PG&E established the following deflector performance requirements:

1 - Deflect the power jet from the runner in less than 2 seconds to reduce speed rise of generating unit to less than 30% above synchronous speed. Close needle at a rate which limits penstock pressure rise to less than 10% above static pressure.
2 - Withstand full jet thrust at any deflector position.
3 - Retract deflector from full jet from any position.
4 - Automatically insert deflector into full jet upon loss of governor oil pressure (fail safe system).

Pelton turbine jet deflectors are typically furnished by the turbine supplier and designed based on experience and/or experimental results. PG&E tried unsuccessfully for three years to get various turbine suppliers to consider retrofitting deflectors onto turbines at Caribou 1 Powerhouse. In 1987, PG&E proceeded in-house with the detailed design of the deflector system. We were fortunate in having the assistance of Mr. Percy B. Dawson, former Chief Mechanical Engineer of the Pelton Water Wheel Company and two other Senior Mechanical Engineers also from Pelton Company to help us with the design. Stirrup type deflectors were selected as being most representative of the Pelton Company experience.

Layout and fabrication drawings were prepared for each powerhouse with particular attention being paid to avoiding interference problems with other components and to provide optimum orientation of the deflected jet. Typical deflector plan and sections are shown in Figures 1, 2 and 3.

After the feasibility level design was complete, a powerhouse outage was taken. The purpose of this outage was to verify the location of the jet deflector support and the critical clearances between deflector and the runner, and the clearance between deflector and the needle tip with deflector traveled from fully retracted to fully inserted position. The wooden jigs have been used to check these clearances and location within the turbine pits. In addition, during this outage a constructability review was conducted with input from both operating and construction personnel.

All deflectors are of cast martensitic stainless steel, ASTM A743, CA6NM for it's higher strength and it's higher cavitation/erosion/corrosion resistance. The deflector

and assemblies were fabricated and machined in a local shop which allowed for close coordination between design and fabrication.

STRUCTURAL / CIVIL: During the feasibility outage, the dimensions of the turbine pit were measured, the powerhouse foundation was detailed, concrete cores were taken, concrete anomalies and construction joints were located, and the relationships between the piping and mechanical equipment were detailed. The existing location of bearings and equipment supports was checked. Care was taken so that any proposed modifications would not adversely affect the foundation conditions. The following were determined:

1. The condition and adequacy of existing equipment and their foundations/supports.
2. The geometry and location and all electrical and mechanical equipment and power cables.
3. The adequacy of the supporting structures and structural materials.
4. Any factors which would effect the long term safety and reliability of the proposed work.
5. Any environmental factor which would change as a result of the deflector system.

It was important to attach the jet deflector assembly to areas free of construction joints or concrete cracking. As a result the exact locations of the anchor bolt holes were field located prior to final design. In this way the bolt pattern on the support plate was adjusted in the final design to more closely fit the existing field conditions.

During the installation, the anchor bolt holes were drilled into the back wall of the turbine pit to provide support for the jet deflector assembly. The deflector assembly was then mounted, plumbed and aligned. The bolt holes and the area behind the base plate was then backfilled with grout. Written grouting and post-tensioning procedures were developed to assure proper anchorage.

As with any installation, proper functioning of the system depends on an integrated design, construction, testing, operations and maintenance effort.

TURBINE PIT AREA: Since the equipment was installed for units with high heads varying from 600 psi to 1100 psi, the jet exit velocity varies from 200 to 350 feet per second . These velocities can remove mill scale from steel liner plates and erode concrete or grout. In general, our practice is to install one-half inch steel plate in the turbine pit area to deflect the jet. The plate is stiff enough to prevent plate vibration during load rejection. The plate liners are bolted in place with an epoxy grout pumped behind the steel plate to assure contact with the concrete surface.

The high velocity jet from the deflector system may be contained within the turbine pit area as in the case of Tule Powerhouse, where by-pass valve had discharged into a pool area upstream of the discharge channel. The deflector was required to discharge into this pool by dissipating most of the jet energy within the turbine pit, this method

prevented the jet from shooting out into the river. In all cases, the magnitude, trajectory and consequences of the deflected jet should be carefully evaluated with a deflector scheme. The environmental impact of the deflected jet must be also evaluated.

SYSTEM COMPONENTS:

The effectiveness in reducing penstock pressures is dependent not only on the new deflector system, but on the remaining system components. Service wear in some governor control systems can result in erratic and non-uniform needle movement. As a result, relatively high pressures can be induced within the penstock. Since only the pressure regulator system was replaced, on some units a new governor operating instruction was devised to prevent pulsing the penstock. In systems where governors have been replaced, in addition to the installation of deflectors, the pressure rises have been significantly reduced for all the test conditions.

Since 1987, PG&E has designed and installed deflectors replacing pre-existing pressure regulator systems in four powerhouses. Table 1 shows a summary of deflector operating data for these powerhouses.

START-UP TESTING

Each unit was tested for several conditions of operation as follows:

1. Open and close each needle in dry state to verify stroke and timing and calibrate test equipment.
2. Insert and retract each deflector to verify position, timing and calibrate test equipment.
3. Pressurize nozzle and open turbine-shut-off valve.
4. Initial unit roll - slightly open each needle and trip deflector and then close needle.
5. Perform speed-no-load, 25%, 50%, 75%, and full load rejections on each wheel. It should be noted that double overhung machines are normally operated under balanced load conditions. Single wheels should be tested only for units that support single wheel operation.
6. Perform multi-unit load rejections.
7. Loss of governor oil test.

Parameters measured during the load rejection operation included:

1. Needle position
2. Deflector position
3. Penstock pressure
4. Unit speed
5. Deflector servo-motor pressure
6. Turbine-generator output

TABLE 1 - SUMMARY OF DEFLECTOR OPERATING DATA

POWERHOUSE DATA	CARIBOU 1 PH	A. G. WISHON PH	BALCH 1 PH	TULE PH
Turbine units (all units horizontal Pelton type)	Three Allis Chalmers Co. double overhung	Four Doble Co. single overhung	One Allis Chalmers Co. double overhung	Two Allis Chalmers Co. single overhung
Date of commission	1921	1910	1927	1914
Maximum gross head (ft)	1,151	1,412	2,379	1,544
Flow (cfs)	1,114	235	213	66
Horsepower per unit	30,000	7,000	40,000	4,500
Original Pressure Regulator	Angle valve	Auxiliary nozzle	Angle valve	Angle valve
Minimum normal governor oil pressure	170 psi	130 psi	150 psi	160 psi
Deflectors installed:	1988 1 - Controlled by single acting servo-motor with spring to close 2 - Deflectors not controlled by governor 3 - Deflectors used to synchronize and emergency only 4 - Uses oil pressure from existing governor system	1989 1 - Controlled by double acting servo-motor plus spring to close on loss of governor oil pressure 2 - Deflector controlled by existing governor, added position feedback. 3 - Deflector used for governing. 4 - Uses oil pressure from existing governor system	1990 1 - Controlled by double acting servo-motor plus spring to close on loss of governor oil pressure 2 - Deflectors controlled by new governor 3 - Deflectors used for governing 4 - Uses oil pressure from new governor system	1990 1 - Controlled by double acting servo-motor plus spring to close on loss of governor oil pressure 2 - Deflectors controlled by new governor 3 - Deflectors used for governing 4 - Uses oil pressure from new 600 psi governor oil system
Needle control:	Remains by governor, with orifice added to limit needle rate	Installed a new DC motor operator	New governor at existing pressure	
Wheel pit lining:	Partially lined	Lined	Partially lined	Lined
Baffle plates:	Baffle plates added	Baffle plates added	No baffle plates	No baffle plates, jet discharges into turbine pit pool.

Each parameter was recorded as a function of time.

At the Tule Powerhouse, the deflector shape was changed to deflect the jet into the existing turbine pit pool. However the actual hydraulic forces acting on the new deflector exceeded the values used for design. Start-up tests measured the hydraulic forces and modifications were made to permit cutting off the full jet upon loss of governor oil pressure.

The turbine pit was inspected and photographed prior to initiation of testing. After the completion of the test series, the pit was again inspected to insure that no damage had resulted from the high velocity jet to the powerhouse.

The start-up testing also verified the integration of the new equipment with the existing equipment. As an example, for certain deflector positions water was forced through air vents in the turbine housing onto the turbine floor. The turbine air vents had to be modified to supply sufficient air while at the same time preventing turbine water from entering the powerhouse while the deflectors were in place.

The maximum penstock pressure rises and speed rises for different powerhouses are shown in Table 2.

TABLE 2 - SUMMARY OF TESTS RESULTS BEFORE AND AFTER MODIFICATION

	Pressure Rise Before	Pressure Rise After	Speed Rise Before	Speed Rise After
CARIBOU 1 PH	27.3%	9%	56.9%	18%
A.G.WISHON PH	20%	1%	35%	9%
BALCH 1 PH	12%	3%	28%	15%
TULE PH	23%	1.5%	N/A	12%

CONCLUSIONS

The deflector systems installed at four PG&E Hydro Powerhouses have reduced penstock pressure rises and turbine-generator speed rises. Were governors have been replaced in addition to the installation of deflectors, the pressure rises have been significantly reduced for all operating conditions.

The retro-fitting of deflector systems into existing powerhouses requires the coordination, communication, and involvement of all disciplines from an early stage to completion of the project. These stages are preliminary testing, feasibility

study, design, construction and start-up. All of these areas must be integrated with the normal plant operation and maintenance.

FIGURE 1 - JET DEFLECTOR
SECTION B

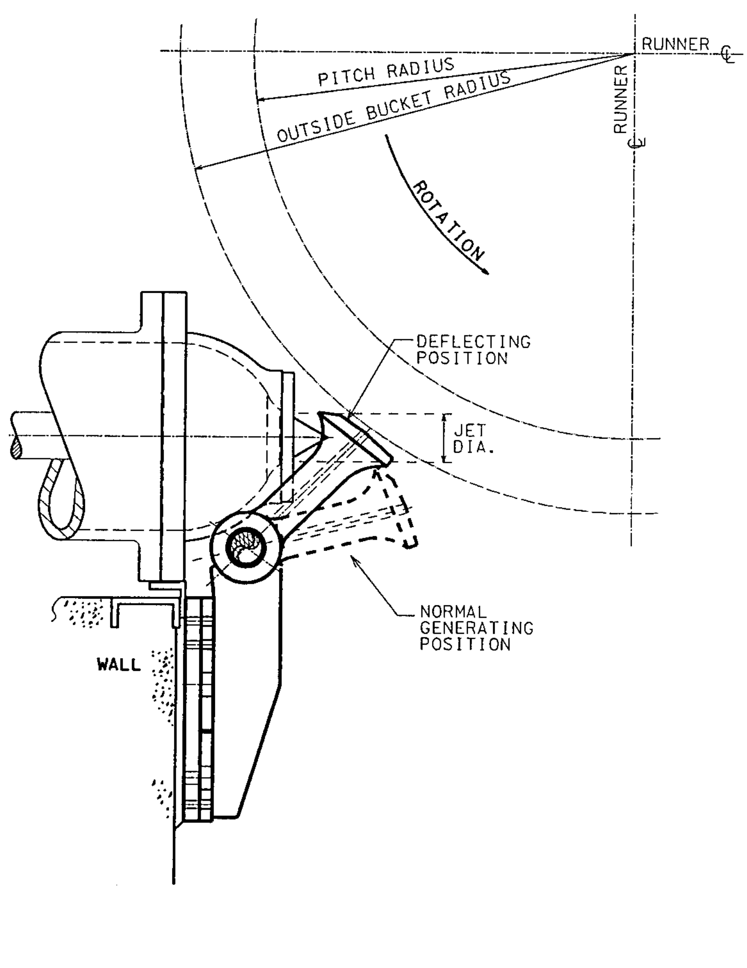


**FIGURE 2 - JET DEFLECTOR
PLAN**

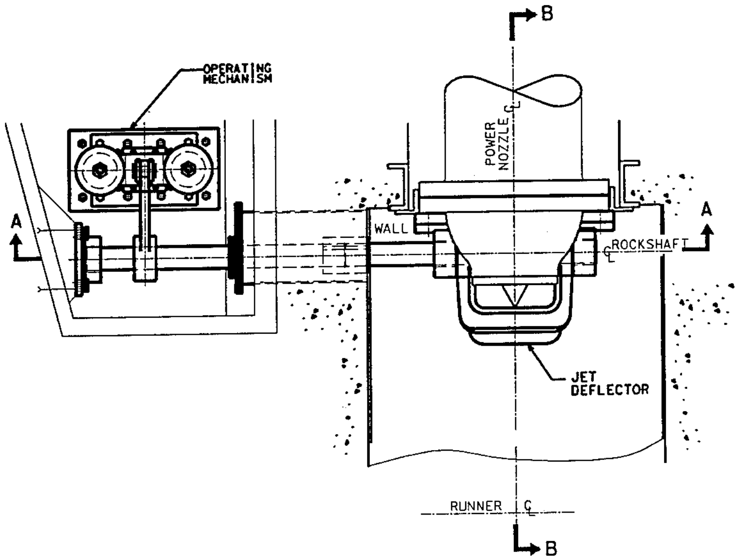
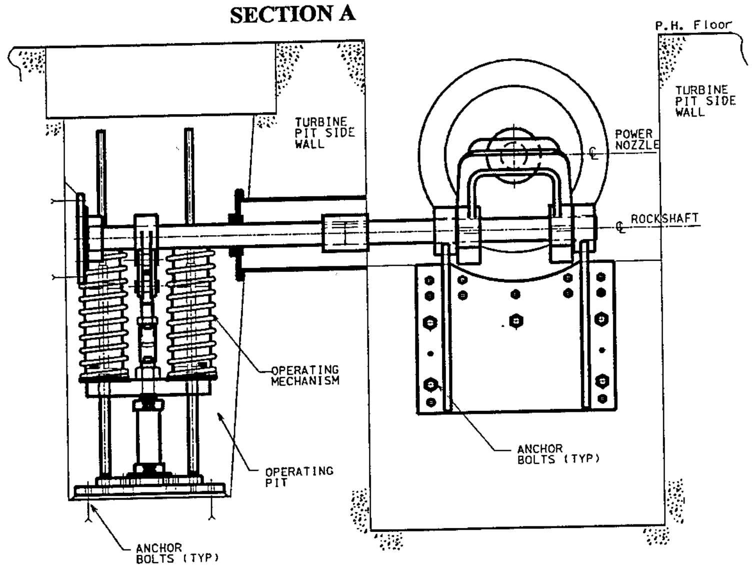


**FIGURE 3 - JET DEFLECTOR
SECTION A**



Replacement Of Great Falls Units 1 & 2
Hydro Turbines

Timothy A. Jablonski P.E. , Member ASCE[1]

Abstract

Great Falls Units 1 & 2, Duke Power's third and fourth hydro turbine replacements, went on line in June of 1991 and June of 1992. The Great Falls facility originally constructed in 1907 consists of 8 hydro turbines each producing 3000 KW of electricity for the Duke Power System. The original Unit 1 & 2 turbines built by Holyoke Manufacturing were 1.22 meter twin horizontal Francis type turbines controlled by cylinder gates. Each unit produced 5200 HP driving a Westinghouse 3750 KVA rated synchronous generator at 225 RPM.

This paper presents the experience gained and lessons learned in rehabilitating hydro turbines in existing civil structures. Specific design challenges included:

1. Maximizing energy output through an improved hydraulic runner design and use of roller bearings.

2. Reducing maintenance by eliminating the wooden turbine shaft center bearing.

3. Installing a new control system based on the use of proportional directional control valves.

4. Automating the units through the use of a programmable logic controller.

1 Design Engineer, Duke Power Company, P. O. Box 1006, Charlotte, NC 28201-1006, (704) 382-9184.

Introduction

In 1981 Unit 1 suffered major damage to the downstream cylinder gate that required the lowering of the headgate to stop the unit. The subsequent turbine inspection found the cylinder gate damaged beyond economical repair. In the process of stopping the turbine the thrust bearing sustained damaged due to unbalanced hydraulic forces. The runner buckets exhibited significant thinning due to corrosion and erosion. The runners had also been weld overlaid numerous times over the years to repair cavitation damage but could possibly withstand one more series of repair welds. Unit-2 had been out of service for several years also due to irreparable turbine damage. After 84 years of service, Units 1 and 2 reached the end of their economic service life and Unit rehabilitations would be required.

Economic Analysis

With the help of several turbine manufacturers, preliminary quotations were solicited for the equipment costs involved in replacing the hydraulic turbines. The ideal hydraulic design called for upgrading the cylinder gates to wicket gates. While the wicket gate design is a hydraulically superior design a significant amount of concrete excavation would be required to assure unrestricted water flow into the lower portion of the gate cases. On the basis of our preliminary material and labor estimates a decision was made to solicit revised quotations based on a cylinder gate design. Because Duke Power maintains its own construction and maintenance staff, a two part detailed cost estimate was prepared for review. The first part consisted of what we called the "Engineering Only" phase. In this phase the selection of the turbine supplier was determined through competitive evaluation and detailed drawings were submitted. The second or "Construction" phase consisted of a detailed construction estimate based on the drawings supplied by the turbine manufacturer. The final cost-benefit analysis showed that over the next forty years it would benefit Duke Power more to rehabilitate the two Units using cylinder gates rather then retire them.

Vendor Site Visit

During the Economic Analysis phase, quotations were solicited from several turbine suppliers. As part of the quotation preparation, site visits were made by each vendor. During the first visit it was discovered that the upstream and downstream bulkheads were out of plum by as much as 12.7 mm. The draft tube flanges were also found to have shifted and would require on site machining to correct the perpendicularity of the flanges in relation to the shaft centerline.

Vendor Scope Of Supply

Through competitive evaluation, Voith Hydro Incorporated of York, PA was chosen as the turbine supplier. Voith's scope of supply included new fabricated runners, turbine shafts, stainless steel shaft sleeves, head covers, stuffing boxes, glands, cylinder gates, stay rings, thrust and guide bearings, hydraulic operating cylinders and the design for the new control system. Also included in Voith's scope would be the fabrication of an adapter ring to correct the plum on the Unit 2 downstream bulkhead. The existing camel backs were shipped to Voith's York facility where the flange faces were remachined to match the new stay rings. Split shims were supplied to correct for metal removed during the machining operations. Voith's Field Services were contracted to line bore the turbine to generator coupling bolt holes.

Powerhouse Preparation

During the turbine manufacturing time, several powerhouse modifications were completed. The penstock liners were sandblasted, their seams sealed and the entire liner coated with coal tar epoxy. PCI Energy Services of Jonesboro, Georgia was then contracted to field machine the draft tube and bulkhead flanges.

Machining the draft tube flange required first establishing the correct setting or elevation of the turbine. To establish the setting, a Duke Power survey crew used the generator guide bearings to sight a laser beam that was directed through the Unit. Monuments were then attached to the penstock wall to give the machining crew a point to measure from.

The machining set up for the horizontal draft tube flanges consisted of centering two spiders inside the draft tube. A cutting bar with drive motor was then attached and the face machined perpendicular to the turbine-generator shaft.

The machining set up for the vertical bulkhead flanges required fabricating a rectangular I-beam frame and then clamping it to the remachined draft tube flange. One spider was then placed inside the I-beam while another was placed inside the bulkhead flange. Alignment was again accomplished by using the laser beam directed through the generator guide bearings. The Unit 1 upstream and downstream and Unit 2 upstream bulkheads were machined using this set up.

The Unit 2 downstream bulkhead machining set up was more complex. Here the flange face was so far out of plum that machining would remove too much metal resulting in a structurally weak flange. To avoid this, an adapter ring 2.46 meters outside diameter and 2.28 meters inside diameter by 44.4 mm thick was fabricated and bolted to the

bulkhead before machining. This adapter ring was attached using 48, 1.0 inch high strength Ferry Cap screws counter sunk into the adapter ring. To prevent possible water leakage into the powerhouse an o-ring was installed between the adapter and bulkhead. The adapter ring was then machined plum.

Turbine Installation

Setting and aligning the camel back draft chest was the basis of the turbine assembly. Since the camel back and draft tube flange faces were machined, a spacer shim was required to bring the camel back up to the proper elevation of the generator center line. After aligning the camel back with jack screws shim thickness measurements were taken and forwarded to Voith for the fabrication of the split shim. Final dimensions were 2.84 meters O.D. by 2.59 meters I.D. by 7.59 mm thick for Unit 1 and 8.00 mm thick for Unit 2.

To aid in the installation of the shim, slots were cut into the shim where the jack screws were located. This allowed the insertion of the shim without removing the jack screws or altering the alignment. After the shim was installed and the flanges torqued down a final check of the alignment was made. While the alignment was within specifications, feeler gauges showed that there were several places where the flange did not seal completely. These gaps if not sealed would result in water continuously bypassing the runners and emptying directly into the draft tube. After several unsuccessful attempts at hand grinding the shims it was decided that the shim would be removed and polymer shim cast in-place. Having had some previous experience with the polymer Duraquartz, Duke contacted the local representative for the supplier Palmer International, Inc. Duraquartz, which is a three part polymer, has virtually zero shrinkage, bonds to metal surfaces, is non-porous water repellent and has a high compressive strength. After evaluating the product and its turbine applications, it was decided that Duke would use the Duraquartz with one modification. To minimize any compressive stresses in the polymer shim, small sections of the Voith split shims were installed at several locations around the flange circumference. The Duraquartz shim was then constructed by gluing a strip of gasket material to the inside diameter of the flanges and trowelling the Duraquartz in between the flange faces.

Stay Rings and Runner

The stay rings that serve to direct the flow of water into the runners also serve as the structural members joining the camel back to the

penstock side of the headcovers. The stay rings were designed as split rings so the shaft and runners could be installed as one assembly. This simplified the installation and reduced the labor and the length of the outage.

To maximize the energy output of the new turbines, the diameter of the runners increased from 1.22 meters to 1.67 meters. This combined with the vendor's improved hydraulic design and improved manufacturing processes increased the horsepower from 5200 to 5600 hp at 21.64 meters gross head. Material for the fabricated runners consists of carbon steel crowns, stainless type 304L bands and cast CA6NM buckets.

Headcovers

Headcovers for the new turbines measure 2.46 meters outside diameter. These headcovers, designed to bolt to the existing bulkheads proved to be a minor source of trouble requiring more labor to install them than was originally estimated. The bulkheads originally installed in 1907 did not have an accurately drilled pattern of holes for the studs. This resulted in interferences that could not be cleared by the new bulkhead bolt hole tolerances. To correct this three of the new headcovers had to have the bolt holes slotted. By using a magnetic based drill the holes on the Unit 1 downstream and Unit 2 upstream headcovers were slotted to fit the stud pattern in the existing bulkheads. The Unit 1 upstream bulkhead having up to 12.7 mm of interference required the most rework. Here, in addition to slotting the bolt holes, the surface of each hole was respotfaced because the nuts were shifted to far to fit in the factory spotfaces. Once respotfaced new oversized washers were fabricated and spot welded to the headcover. After installation all headcovers were recoated. The Unit 2 downstream headcover that initially was estimated to require the most labor to install turned out to be the easiest. On Unit-2, an adapter ring was bolted to the downstream bulkhead and then the headcover was bolted to the adapter ring. Since both bolt circles matched perfectly the installation went smoothly.

Shafting System and Bearings

With reduced maintenance being a primary objective, the design specifications for the replacement turbine required the elimination of the center wood bearing located inside the camel back. To meet this specification Voith designed a shafting system with a 406.4 mm O.D and 254.0 mm I.D. hollow shaft. The span between bearing centers is 8.09 meters. To complete the shafting system a carbon steel stub shaft is bolted to each runner crown. The complete rotating assembly is supported by a 342.9 mm SKF spherical roller combination thrust and guide bearing and 355.6 mm Cooper split roller guide bearing. The final

coupling of the turbine to generator shaft uses standard flanges that were line bored by Voith.

Turbine Hydraulic Control System

The design of the hydraulic control system was based on the synchronization of the two cylinder gates when opening or closing. To minimize any thrust load on the thrust bearing, the control system was designed to keep the relative position of the two cylinder gates within 10% of each other. To accomplish this the system uses two Bosch proportional directional control valves, two rotary potentiometers and a Square D Programmable Logic Controller. The downstream gate was designated as the lead gate and the upstream gate the slave gate.

When the gates open a feedback signal from each gate position transmitter is compared in the PLC and an error signal sent to the lead proportional valve to either hold or continue the movement of the lead gate. In addition to the normal opening or closing functions, an emergency closure circuit was incorporated in the hydraulic flow circuit. In the emergency close mode, hydraulic oil from the accumulator directly flows into the hydraulic cylinders through Racine directional control valves bypassing the Bosch proportional control valve. To control the speed of the system Racine needle valves are utilized in the discharge line of the hydraulic cylinders. To reduce piping and avoid oil leaks all valves are stacked on aluminum manifold blocks. System design pressure is limited to 300 psi so valve sizing was a compromise between size and pressure drop. At Great Falls each unit has four hydraulic cylinders that require 57.65 liters of oil total for one gate movement.

To complement the control system, a new hydraulic power supply was designed and installed by Duke Power. The hydraulic power unit consists of two 100% capacity Brown and Sharpe rotary gear pumps, a Pall duplex hydraulic oil filter and 1419.5 liter accumulator tank. Size of the accumulator was based on having enough oil pressure to shut down Units 1 & 2 and still have enough reserve pressure to restart one of the Units.

Turbine Automation

The replacement of the Unit 1 and Unit 2 hydraulic control system with independently controlled cylinder gates required precise gate position monitoring and control. A programmable logic controller was selected to provide the necessary analog outputs to control the Bosch proportional valves and also provide a large degree of control functions and automation.

To provide automatic start-up and shut-down the gate position, speed of movement, the generator field circuit breakers, generator field rheostats, generator circuit breakers and the turbine driven exciters were automated. To control and automate the various pieces of equipment, it was necessary to monitor turbine gate position, turbine bearing temperatures, generator bus frequency and voltage, system frequency and generator to bus phase angle. Synchronizing was accomplished by comparing the system phase angle to the generator bus phase angle. To maintain proper voltage to the generator fields while synchronizing, the PLC monitored the exciter output voltages and adjusted the exciter turbine wicket gates to maintain proper voltage output. The sequence of steps programmed into the PLC consist of the following:

Step 1 Open gates to approximately 10 % gate position.

Step 2 Close generator field circuit breaker.

Step 3 Raise generator field rheostat to adjust field amps and adjust exciter gates to load follow.

Step 4 Begin adjusting gate position to raise turbine speed to synchronous speed.

Step 5 Adjust field rheostat (field amps) to match generator voltage to system voltage.

Step 6 Compare and match generator voltage phase angle to system voltage phase angle by jogging turbine speed up or down, when in sync initiate generator circuit breaker close signal.

Step 7 Open gates to 100% gate position.

Step 8 Adjust field rheostat to adjust power factor.

Step 9 Begin monitoring sequence and trip generator if an abnormal condition occurs. Items to be monitored are: turbine guide and thrust bearing temperature, hydraulic system pressure gate position imbalance, excitation high or low and too many attempts to auto synchronize.

Conclusion

Small turbine rehabilitation offers the hydro electric industry a means of maximizing energy production. At Great Falls a hydraulically improved turbine coupled to an upgraded Class F insulated generator increased energy output from 3.0 to 4.0 MW. Overall turbine maintenance was reduced while availability and reliability increased. For relatively small capital investments, Great Falls Units 1 & 2 were returned to service with an expected 40 year operational life.

Upgrade of the Chippewa Falls Hydroelectric Turbines

Mark Holmberg[1]
Bill Zawacki[2]
Donald R. Froehlich[3]
Jim Singleton[4]

Abstract

The six 3.7 MW vertical manual Kaplan turbines in the 65 year old Chippewa Falls powerhouse will be upgraded by replacing the turbine runners. This paper describes the technical considerations and procurement approach used to achieve the expected increase in turbine performance. Since homologous model tests are too costly and cannot be justified for this project, the techniques used to evaluate alternative performance guarantees are also presented.

Introduction

The 21.6 MW Chippewa Falls hydroelectric plant on the Chippewa River in Chippewa Falls, Wisconsin, was placed in commercial operation in 1928. The plant is operated by Northern States Power, in conjunction with five other hydro plants on the Chippewa River, to provide daily peaking power to its system.

[1]Project Manager, Northern States Power, 414 Nicollet Mall, Ren. Sq. 7, Minneapolis, Minnesota 55401.

[2]Manager, Wisconsin Hydro Plants, Northern States Power, 100 North Barstow, Eau Claire, Wisconsin 54702.

[3]Partner & Project Manager, Black & Veatch, PO Box 8405, Kansas City, Missouri 64114.

[4]Mechanical Engineer, Black & Veatch, PO Box 8405, Kansas City, Missouri 64114

The Chippewa Falls Plant is normally operated at best efficiency in combination with the six unit 36.8 MW Wissota Hydro Plant located 3 miles upstream, as there is no effective storage between the two plants. Efficiency tests using the current meter method determined the best efficiency of the Chippewa Falls and Wissota turbine generator units to be 76 percent and 81 percent respectively. The efficiency tests also determined the Wissota maximum discharge and best efficiency point with all units in operation to be 10,600 cfs and 9,480 cfs. The feasibility of expanding the Wissota Plant was subsequently investigated, but expansion was not economically viable. Furthermore, since the 12,000 cfs maximum plant discharge at Chippewa Falls is greater than the Wissota Plant discharge, expanding or increasing the Chippewa Falls plant was also not justified. Rehabilitation alternatives for the Chippewa Falls units were evaluated and replacing the runners, which would significantly increase turbine efficiency, was found to be most cost effective.

Future Chippewa Falls minimum plant releases will be 1,000 cfs from April 15 through May 30, and 785 cfs for the remainder of the year. Therefore, at least one turbine will be rehabilitated as a manual Kaplan turbine to provide flexibility to operate at the minimum release conditions as well as full load when flows permit. A second turbine will also be rehabilitated as a Kaplan turbine to increase operating reliability. The other four turbines will be rehabilitated as less costly, fixed blade propeller turbines since the plant will be operated as a peaking plant in combination with the upstream Wissota Plant.

	Wissota Plant Discharge	Chippewa Falls
Best Efficiency	9,480 cfs	5 Units @ 1,900 cfs
Maximum Capacity	10,600 cfs	5 Units @ 1,920 cfs and 1 Unit @ 1,000 cfs

Turbine Generator Units

The turbines, rated at 5,000 hp at 29.6 ft net head, were manufactured by Allis Chalmers in 1928 and have never been disassembled or rehabilitated. The Chippewa Falls turbines are unique because they are vertical manual Kaplan turbines and have a symmetrical semi-spiral intake and low Moody draft tube. Cross sections of the powerhouse and turbines are shown on Figures 1 and 2, respectively. The Kaplan turbine runner blades are manually adjustable from a mechanism at the turbine/generator shaft coupling. The runner blade mechanism in the hub is grease lubricated; however, the hub does not have seals to retain the grease. Over the years, the grease has continuously washed out and

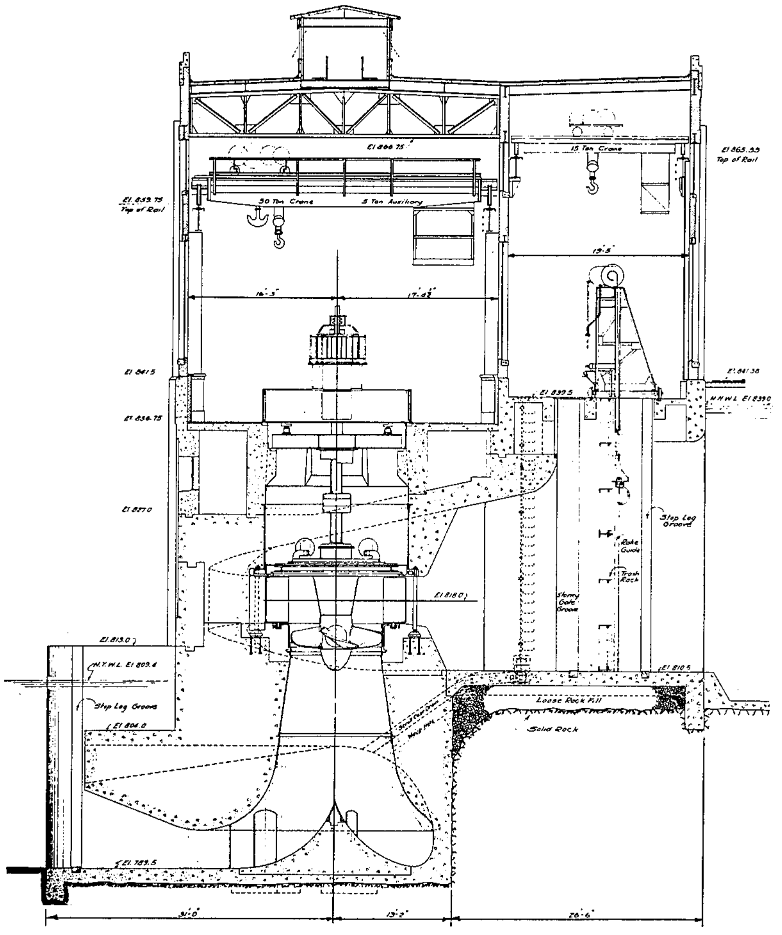


Figure 1. Powerhouse Cross-section

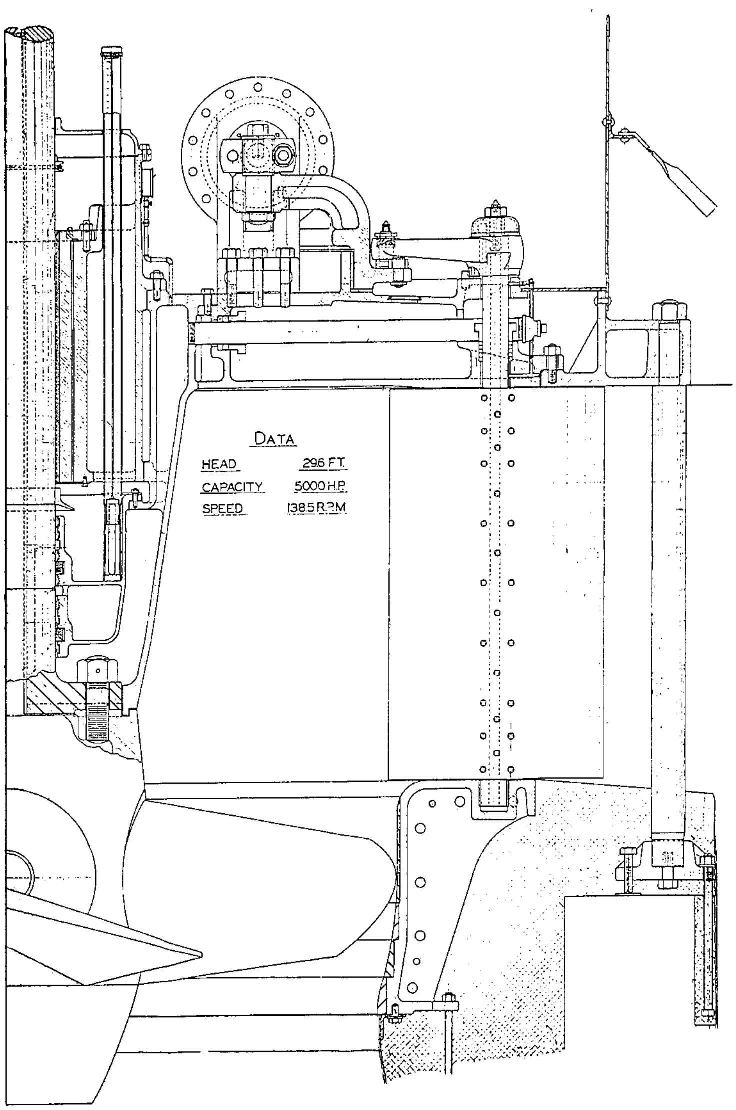


Figure 2. Turbine Cross-section

the runner blade adjusting mechanism has corroded. Deterioration now limits blade movement on most units.

The turbines are in poor condition and in need of major rehabilitation. Significant cavitation repairs to the runner and discharge rings have been necessary on a continuing basis over the years, caused in part by the runner centerline being approximately four feet above the average tailwater level. The turbines have cast iron combination bottom rings and discharge rings which are embedded in the powerhouse concrete, as shown on Figure 2. The discharge rings have replacement wearing rings which can be replaced and remachined. In the past, cavitation damage on some units required replacement of portions of the replaceable discharge ring, wearing rings, embedded discharge ring, draft tube liner, and concrete. Past repairs were made without disassembling the unit.

Inspection of the unwatered turbines revealed that the clearance between the runner and discharge ring was about 0.5 inch, compared to the original design of 0.2 inch. Repeated cavitation repairs to the runner and discharge rings distorted the original surface profiles. The necessary major repairs and remachining of the runner and discharge ring would require disassembly of the turbine unit.

The air cooled generators were manufactured by General Electric and are rated at 4,500 kVA at 0.80 power factor. The generators were rewound between 1982 and 1988 and are in good condition. Therefore, rehabilitation of the generator will not be required.

Turbine Performance

Modern turbines are designed to maximize performance for site specific operating head and tailwater conditions. The turbine runner design is optimized in coordination with the design of the other turbine hydraulic components, including the turbine intake, stay ring, wicket gates, discharge ring, and draft tube. Modern turbine designs have evolved over the years to improve turbine efficiency and operation. However, newer turbine designs have dramatically different water passage, runner, and turbine dimensions.

Current turbine and runner hydraulic designs are commonly developed with the aid of three dimensional hydraulic computer models. The computer design tools allow the inexpensive evaluation of alternative designs to optimize the turbine and runner design. Computer models are the most accurate in predicting turbine performance near the best turbine efficiency; however, model accuracy is reduced in off peak areas away from the most efficient head and flow.

Since hydraulic computer models cannot predict turbine performance with the accuracy required to confirm performance guarantees, turbine designs are usually verified by reduced scale physical model tests. Suppliers commonly design more than one runner for testing in a hydraulic

laboratory. The final runner designs are refined during the physical model tests that provide accurate performance and cavitation test results and allow visual observation of cavitation development on the runner.

Homologous model tests provide the best technical assurance that the prototype performance will be achieved. The International Electrotechnical Commission (IEC) Model Test Code has established hydraulic similitude relationships as well as dimensional proportions and manufacturing tolerances necessary to achieve the prototype performance. Homologous model tests results are often accepted as confirmation that the turbine design will meet the performance guarantees. The costs for a homologous model test range from $300,000 to $750,000, depending on the type of turbine and model tests required. Unfortunately, homologous model tests to confirm performance are not always justified for runner replacements unless there are many large runners or large capacity units to support the expense.

Designs for replacement runners are often based on modern model runner designs which have similar operating conditions and turbine dimensions. The replacement runner performance is predicted by extrapolating and adjusting the existing model data for differences between the turbine hydraulic dimensions. The efficiency gains of modern turbines have been achieved by the combined design improvement of all hydraulic components. Since it is often impractical to significantly modify or replace other turbine components, the replacement model performance is determined by reducing the modern model runner performance to account for any differences between the model and turbine, as shown on Figure 3. The most significant turbine hydraulic water passage differences that affect turbine performance are the wicket gate height, wicket gate pin circle diameter, stay vanes, intake, draft tube, and runner centerline elevation. The amount of any increase or decrease in turbine efficiency can be estimated by either theoretical hydraulic head loss calculations, or preferably, by previous model tests of alternative designs.

<u>Replacement Runner Procurement</u>

A successful turbine runner replacement and rehabilitation program should be tailored for the specific project and turbine condition. The program should also consider the sources for the supply of equipment and work to be performed. Rehabilitation of the Chippewa Falls turbines will require the unit to be disassembled and reassembled, the turbine runner replaced, and the turbine components rehabilitated.

Turbine replacement runners could be obtained from the 8 to 10 major turbine suppliers. However, only certain turbine manufacturers have personnel and facilities in the United States to provide field assembly/disassembly and component rehabilitation services. Runner size and the number of units also influence the number of potential suppliers.

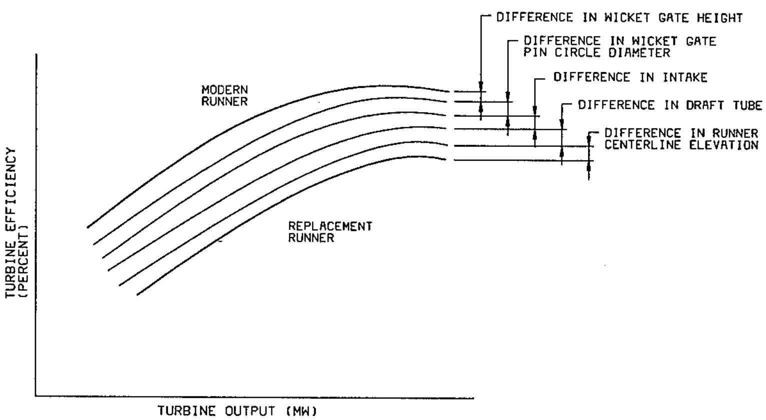


Figure 3. Determining Replacement Runner Performance

Five turbine manufacturers had previous experience and expressed an interest in supplying the six 10 feet diameter Chippewa Falls replacement runners. The turbine runners were procured under a separate contract to foster competition and maximize the number of bidders. Contracts for the unit disassembly and reassembly and component rehabilitation will be bid and awarded separately as well.

The powerhouse electrical equipment and wiring will be rehabilitated concurrently with the turbine work. Separate supply specifications will procure the replacement transformers, switchgear, static exciters, and relay panels while a general construction contract will be prepared for the electrical powerhouse rehabilitation and equipment installation work. The electrical supply and rehabilitation work was not included in the turbine contract since this work would have been subcontracted by all the turbine manufacturers.

Base bids for replacement runners designed for 1,900 cfs at 28.7 ft without modifying the other turbine components were obtained. Alternative bids allowing modifications to the discharge ring and other

turbine components were also requested if the bidders believed the cost of the turbine modifications would be justified by the efficiency gains. The value of unit efficiency was provided to the bidders to assess the potential for any proposed modifications.

Replacing the discharge ring could increase the turbine efficiency by increasing the runner diameter and lowering the runner centerline. Removal of the existing discharge ring, as well as the installation of the new discharge ring, would be necessary. Replacing the existing wearing rings would require field disassembly and "in place" field machining. Therefore, the costs for replacement wearing rings were requested for comparison of any alternative bids. The costs for homologous model tests were also obtained; however, the high cost could not be justified for the capacity of the Chippewa Falls units.

All manufacturers submitted technically sound and commercially competitive bids for runner replacement. However, the guaranteed turbine performance did vary. The model data and technical basis for estimating the replacement runner performance was reviewed to evaluate the proposed performance of all bidders on a common basis. The bidders' model dimensions were compared to the Chippewa Falls turbines and the adjustments for any differences were also evaluated on a common basis.

In comparison to modern turbine designs, the Chippewa Falls turbines do not have stay vanes, but have a smaller wicket gate pin circle, smaller distance between the runner centerline and distributor centerline, higher runner setting, and a Moody draft tube. The most significant of the factors affecting performance are the high setting and Moody draft tube. A comparison of the modern runner model performance would place all bidders on a common basis, but would not evaluate the setting and Moody draft tube effects on the specific runner designs.

Most turbine bidders had previous experience designing replacement runners for similar spreading type draft tube applications and had model tests of the effects of draft tubes similar to Chippewa Falls. The design experience of the bidders also varied since the runner and hydraulic turbine designs of the bidders were derived independently. Estimating the effects of the Moody draft tube also concerned the Bidders when proposing the guaranteed performance. As a result, the most competitive bidders planned to perform limited model tests of the replacement runner, modern wheel case, and Moody draft tube to confirm the proposed design and performance before manufacturing the prototype runners. Therefore, selecting the turbine runner design was made with the knowledge that the replacement runner would be designed and the guaranteed performance shown on Figure 4 would be partially confirmed by a similar if not homologous model test. The model tests would also confirm the blade angle for the propeller turbines and the cavitation margin.

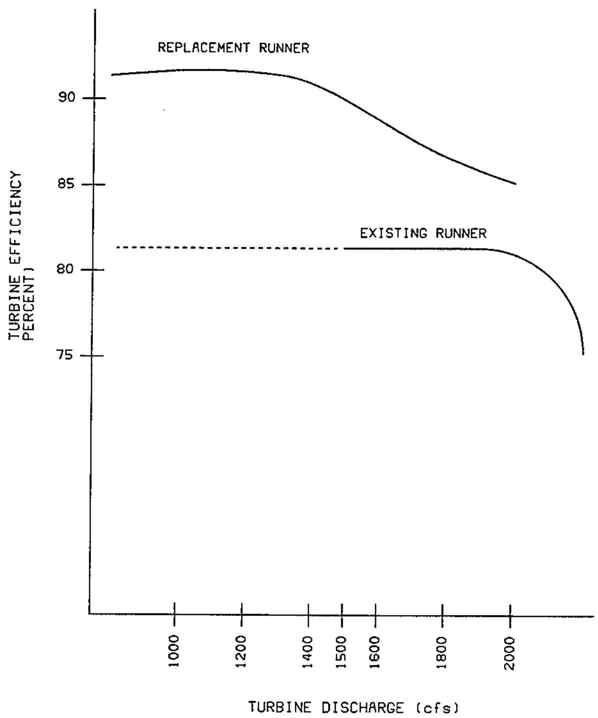


Figure 4. Comparison of Turbine Performance

Further design assurances will be achieved by delivering , installing, and testing the two Kaplan runners first. Field index and capacity tests of the Kaplan runners can confirm the prototype capacity and best blade angle for the propeller turbines before they are manufactured. The replacement runners will be inspected following manufacture to confirm that the hydraulic dimensions are within the manufacturing tolerances required by the IEC Test Code. Therefore, the upgraded performance of the replacement runner for the Chippewa Falls turbines can be expected to be achieved.

The contract for the supply of the Chippewa Falls runners was awarded in January 1993. Rehabilitation of the turbines and powerhouse will begin in January 1994 and be completed in the Spring of 1995.

Three Case Studies of Modifications To Washington Water Power Dams

John Gibson[1]
John Hamill[1]
Ed Schlect[1]
Thomas Haag[2]

Abstract

Washington Water Power (WWP) has an ongoing program to modify dams at their hydroelectric plants in response to the requirements of the recently completed FERC Part 12 Inspection and the need to improve the reliability and operational capability of the projects. This paper reviews the unique approaches taken to three modification projects recently completed by WWP.

Introduction

WWP operates nine hydroelectric plants with a total capacity of 946 MW on the Spokane, Clarks Fork, and Colvill Rivers in eastern Washington and northern Idaho. The dams at two of these plants, Long Lake and Post Falls, have been the subject of modification projects as a result of the previously performed Part 12 Inspections.

The 72.5 MW Long Lake Hydroelectric Development was placed in service in 1915 and is located approximately 25 miles (40 km) northwest of Spokane, Washington, in a scenic, narrow canyon area of the Spokane River. Figure 1 shows the main dam at Long Lake, which is a 213 foot (65 m) high gravity dam with a crest length of 593 feet (181 m). The dam

[1]Project Engineers, Washington Water Power Company, 1411 Mission, Spokane, Washington 99220

[2]Project Manager, Black & Veatch, 11401 Lamar, Overland Park, Kansas 66211

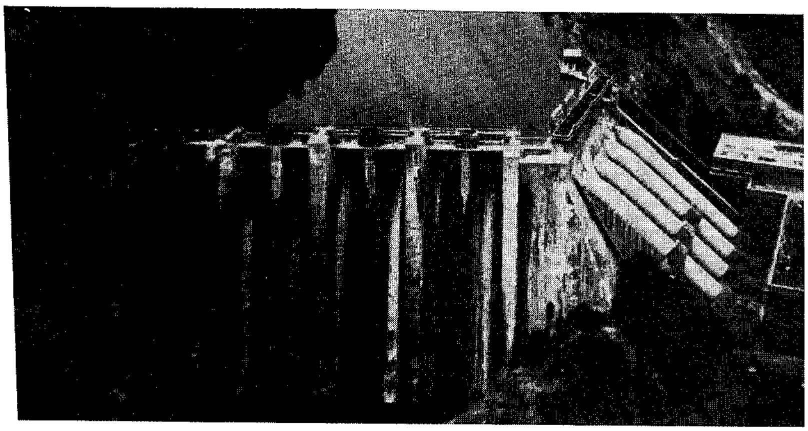


Figure 1. Long Lake Hydroelectric Project

includes a 353 foot (127.5 m) long spillway containing eight vertical gates with a total spill capacity of 115,000 cfs.

At Long Lake, the FERC Part 12 Consultant (Ebasco), supported by underwater survey data, determined that there was a sufficient amount of erosion in the river bed near the toe of the spillway dam and that structural modifications to reduce erosion should be considered. It was also noted that the eight spillway gates were aging and operational difficulties were common, which resulted in reduced reliability.

Units 1 through 5 at the 18 MW Post Falls Hydroelectric Development (HED) were placed in service between 1906 and 1908, with Unit 6 added in 1980. Post Falls is located on the Spokane River, approximately nine miles downstream of Coeur d'Alene Lake, one of the most scenic and popular recreational areas in the Northwest. The Post Falls HED consists of three dams positioned on three separate channels of the river.

The dams are referred to as the North, Middle, and South Channel Dams, and each uses a gravity type dam. The North Channel Dam is 431 feet (131 m) long, 31 feet (9.46 m) high, and contains a gated spillway. The 175 foot (53.3 m) long, 64 foot (19.5 m) high, Middle Channel Dam is integral with the powerhouse, which contains six horizontal Francis turbines. The South Channel Dam is 127 feet (38.7 m) long, has a maximum height of 25 feet (7.63 m), and contains six sluice gates. Figures 2 and 3 show the Middle and South Channel Dams, respectively.

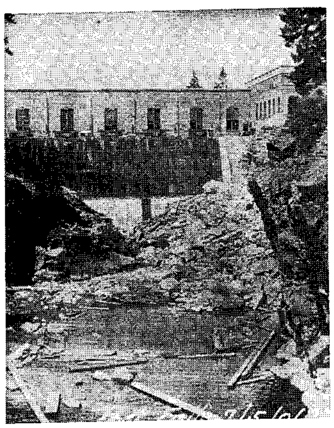


Figure 2. Middle Channel Dam and Powerhouse

Figure 3. South Channel Dam

Ebasco determined that the Middle and South Channel Dams did not meet the required stability factors of safety for Probable Maximum Flood (PMF) conditions.

Case Study 1: Long Lake Spillway Plunge Pool

The Long Lake spillway is integral with the 213 foot (65 m) tall gravity dam. Eight vertical lift gates atop the dam are operated to spill up to 115,000 cfs (3,260 cms) to the tailwater area below the dam. As it spills, the water follows the ogee shape to a plunge pool which was excavated in the bedrock. The original dam design did not include a flip bucket, and the water entered the plunge pool directly at the toe of the dam. WWP monitored the erosion at the toe on a periodic basis using an underwater survey team. The results of these ongoing surveys indicated that the toe of the dam was being undermined, but the data was not conclusive as to the extent, with erosion depths varying significantly between surveys.

WWP installed a flip bucket to reduce the erosion at the toe of the spillway, as shown on Figure 4. The estimated construction cost for the flipbucket and erosion repairs was $5,000,000. Given the high cost of the project and unknown factors related to the ability to dewater and repair the plunge pool, the project team focused on an approach that would reduce the contingencies involved.

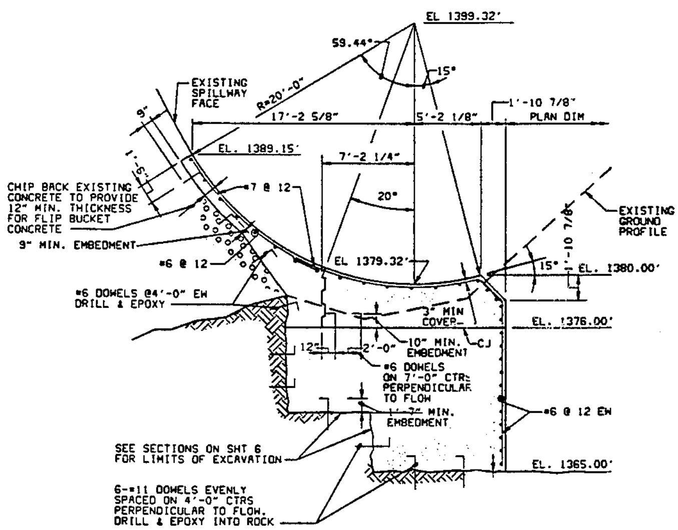


Figure 4. Proposed Flip Bucket Addition

The spillway discharge channel was generally submerged by the backwater from the powerhouse tailwater. However, the channel reach approximately 400 feet (122 m) downstream of the spillway, rose within 7 to 8 feet (2 m) of the normal tailwater level. A removable timber flashboard type cofferdam was constructed and positioned across the channel to dewater the plunge pool area. This would permit detailed inspection of the erosion and confirm the ability to dewater the area. The cofferdam design, shown on Figure 5, consisted of an anchored concrete sill, removable steel wide flange posts, and timber flashboards that can be slipped between the flanges of the posts. The sill was constructed during a period of low tailwater using a small sandbag cofferdam to allow construction in the dry.

Upon completion of cofferdam construction and dewatering, contractors who were to be invited to bid the final construction work viewed the dewatered channel and plunge pool. During the dewatering, it was evident that the erosion was not as severe as the underwater

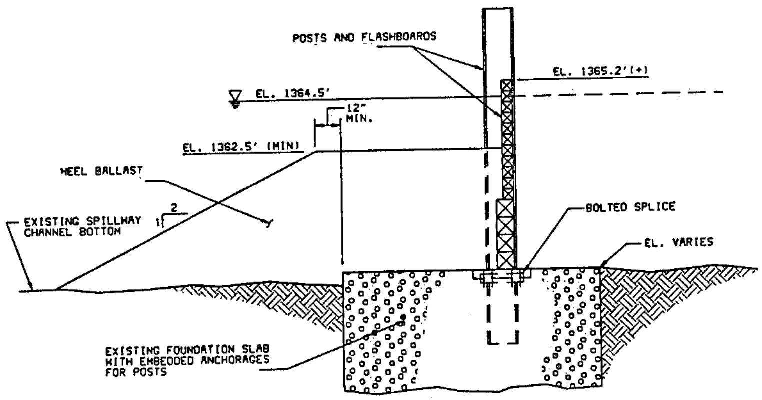


Figure 5. Cofferdam Section and Details

inspections had indicated. After additional stability analysis by Ebasco and consultation with the FERC, it was concluded that no immediate repairs were required and the flip bucket could be deferred.

WWP mapped the erosion in detail by terrestrial photogrammetric methods and installed erosion monitoring reference points. WWP will reinstall the cofferdam, dewater, and remap the area on a periodic basis. The flip bucket hydraulic modelling, detailed design and specifications were completed so that if conditions warrant, the flip bucket can be installed in a single construction season.

Case Study 2: Long Lake Spillway Gate Replacement

WWP experienced difficulties in maintaining the spillway gates at Long Lake in operable condition. This was especially true of Gates 1 and 2, which were inoperable using normal hoisting and could only be raised using portable jacks. Each of the eight spillway gates were approximately 26 feet (8 m) wide by 29 feet (8.8 m) tall and were hoisted using an electric motor driven, rack and pinion drive system. The problems with the gates were related mainly to an increase in reservoir level and corrosion of various components of the gate leaf and hoist equipment.

In early 1991, WWP commissioned Black & Veatch (B&V) to evaluate gate replacement alternatives and design the selected option. The results of the study indicated that wire rope, hoist-operated, fixed wheel roller gates were the best option on the bases of lowest capital cost, gate deck space utilization, and maintainability.

The approach to the design and procurement of the gates was based on the following criteria:

- WWP directly participated in the design effort to permit maximum customization.

- The designer was required to prepare shop drawings to allow participation of local firms in the bid process.

- Time was critical and the approach had to minimize the engineering, bidding, and shop drawing preparation/review effort.

- Planned replacement of the eight gates over a 4-year period, with Gates 1 and 2 to be replaced in 1992.

- The responsibility for the design of the gates needs to reside with the Engineer and not be subcontracted.

To meet these criteria, Black & Veatch designed the gates and prepared the fabrication drawings and details before bidding. The drawings and specification included all piece details, cuts, fits, and assembly information needed to immediately start fabrication. No shop drawings, other than hoist equipment information, were submitted by the fabricator. The design, detailing, bidding, and selection of the fabricator was completed in about six months (March 1992). At that point, gate fabrication immediately began. The gates were delivered and installed by November 1992.

Figures 6 and 7 represent examples of the gate design drawings, with the addition of fabrication information.

Bids were solicited from nationally known gate fabricators as well as qualified local steel fabricating shops. The bids reflected the lower shipping and labor costs that the local fabricating shops could provide. The gate work was awarded to R.A. Hanson of Spokane.

Gates 1 and 2 were fabricated, installed by Goebel Construction Co., and placed in service on schedule in late 1992. Figure 8 shows the

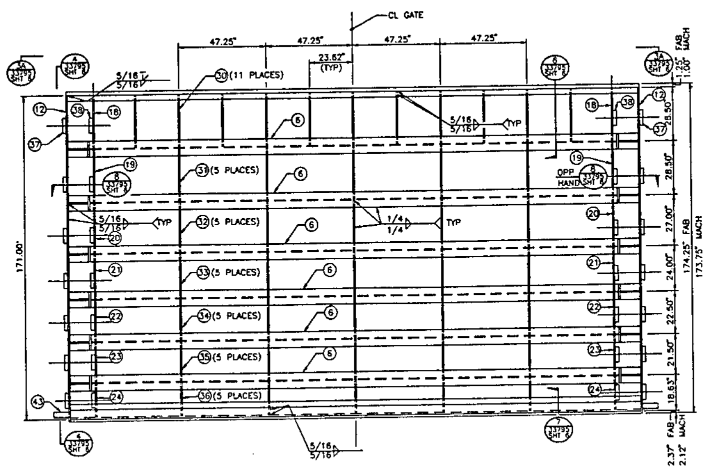


Figure 6. Roller Gate Elevation View

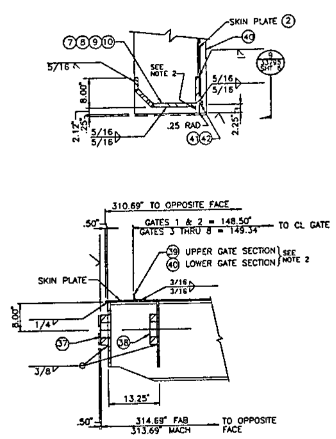


Figure 7. Roller Gate Typical Shop Details

Figure 8. Completed Gate Installation

completed gate installation. The gates are operating successfully with no leakage.

One advantage of this design/detailing approach is that it allows WWP to procure the remaining six gates using an identical design, instead of differing vendor designs which may lead to potential inconsistencies. Gates 3 and 4 are slated for installation in 1993.

This approach has met WWP's criteria of customized design, potential for local shop fabrication, low cost, procurement flexibility, short schedule, and high quality. This approach has also resulted in nearly $300,000 savings in fabrication and installation costs.

Case Study 3: Post Falls Dam Stabilization

As described previously, the Middle and South Channel Dams at the Post Falls Hydroelectric Development were determined by the Part 12 Consultant, Ebasco, to require additional stabilization. WWP hired B&V to evaluate and design alternatives for stabilization with the goals of lowest cost, FERC acceptance, minimal operational impact, no environmental impact, and constructibility.

A modified Value Engineering (VE) Process brought together the combined creative talents and knowledge of the WWP and B&V project staffs as a single team to develop and evaluate stabilization alternatives within a short time frame and at a minimal cost.

The process centered on a two-day workshop, during which the team determined the important project information, agreed to the evaluation criteria, brainstormed and developed alternatives, performed an initial evaluation and ranking, and determined a short list of alternatives

for a more detailed evaluation to be completed by the Engineer. From the start, the comprehensive nature and synergism of the team resulted in a more thorough examination of the problem, a greater range of creative alternatives, and a consensus as to the most likely solutions.

The workshop team brought together personnel from the various areas of concern and expertise, such as environmental and licensing, operations, management, engineering, and project safety. The final aspect which made this process successful was the predetermined structure, using preformated study documents. This forced the team to be efficient and complete in its activities and simplified documentation.

During the first day of the workshop, the team agreed on the baseline project data, studied evaluation criteria, and through the brainstorming process, determined 41 potential stabilization alternatives. The alternatives encompassed various types of anchoring, structural modifications, hydraulic modifications, and engineering only solutions. The second day was spent describing the alternatives in sufficient detail to allow ranking and evaluating to arrive at a short list for final evaluation.

The short list of alternatives was then thoroughly evaluated in the traditional feasibility study approach, and a final selection, post tension anchors, was determined. Given the consensus approach from the beginning, the selection was accepted with ownership by participants and no second guessing.

The post tension anchor solution was designed and implemented during 1992. The Middle Channel Dam was anchored with five, 1645 kip anchors spaced across the intake deck. The anchors are shown on Figure 9. The South Channel Dam was anchored with two 760 kip anchors, one 841 kip anchor, one 420 kip anchor, and two 187 kip anchors. The size of the anchor depended on the specific section of the dam.

The installing contractor was Malcolm Drilling Co., with the anchors provided by Dwidag. The total cost of the work was approximately $900,000 and was completed within WWP's budget in November 1992.

The modified VE approach shortened the study period by approximately three to four weeks. The process assured that the best alternative had been selected for this site and circumstances.

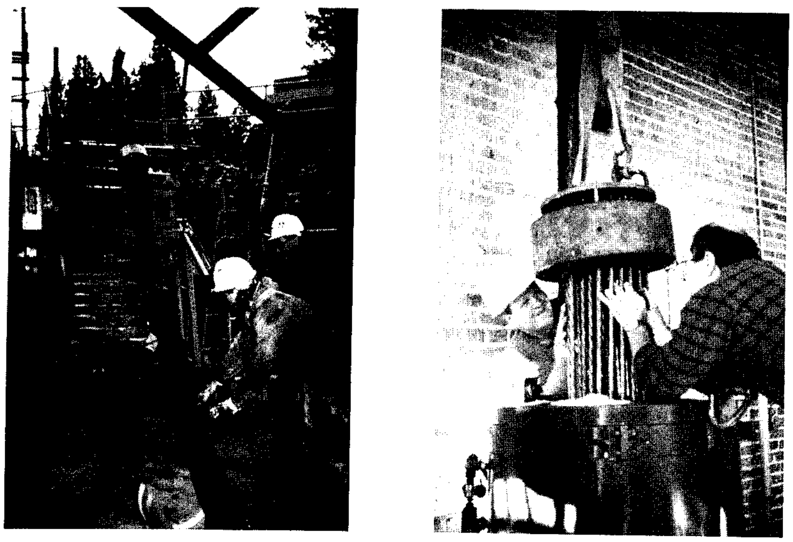


Figure 9. Post Falls 1645 Kip PTA

Conclusions

WWP continually seeks creative solutions and approaches to their engineering and constructions needs. These three cases show that the engineering community can develop and implement alternatives to the traditional solutions.

The teamwork and excellent communication between the WWP and its engineers were also benefits from the three cases. The synergy of combining the talents and experience of both organizations was an important lesson confirmed and one that will be continually used on future WWP projects.

Synchronizing Upgrade for Hydroelectric Plants

by James H. Harlow*

ABSTRACT/INTRODUCTION

Of the various common forms of turbine drive, it is generally accepted that hydroelectric turbines may be the most difficult to bring into synchronism with the power system. The very high mass to kVA ratio found in hydroelectric turbine/generator systems is revealed in the high inertia associated with these machines. Because of the very high machine inertia, the synchronizing control system is especially taxed in its ability to bring the system into synchronism in a reasonable period.

This paper describes the automatic synchronizing system upgrade recently applied to two west coast hydroelectric installations in order to reduce the time required for synchronization and to minimize "bumps" which were evident when using the original synchronizing equipment. The purpose of the project was to replace the existing equipment with equipment of a later design which is better suited to control the turbine speed and to command a more accurate closing of the circuit breaker.

ORIGINAL EQUIPMENT AND INSTALLATION

The existing synchronizing relay was of an early static design where both the circuit breaker closing relay (device 25A), and the speed and voltage control (devices 15 and 90) were accomplished in one package—the package integral to the switchgear. The power source was 125 V dc. Circuit breaker closing involved an interposing circuit breaker closing relay where the resultant closing accuracy of the circuit breaker could be held to only ±10° of the desired electrical angle. The speed matching pulses were of a fixed duration and fixed repetition rate. The raise and lower speed output contacts were wired directly to the governor positioner motor circuit.

Figure 1 depicts the pertinent power and synchronizing control circuit, the "breaker close" lines being wired in series with a sync-check relay to a breaker closing relay. The automatic control output shown, the raise and lower signals for the voltage and speed, may be switched to a manual mode of operation, if desired.

* Beckwith Electric Company, Inc., P.O. Box 2999, Largo, FL 34649-2999.

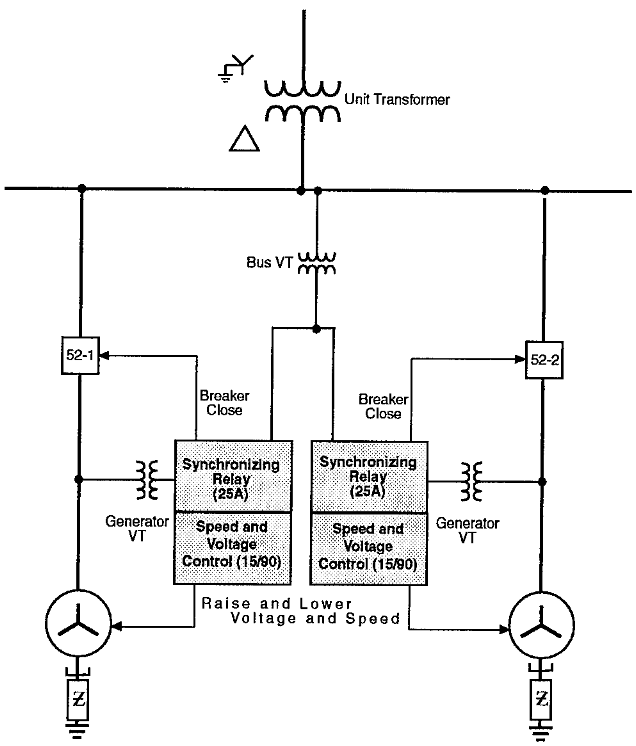


FIGURE 1 System One-Line Diagram with Synchronizing Control

UPGRADE INSTALLATION

The schematic of Figure 1 is unchanged by the upgrade with voltage inputs, the closing circuit and the control output circuit being a simple matter of disconnect-reconnect.

The new relays are powered from a 120 V ac source rather than 125 V dc. The sum of the power circuit burdens of the relays at 120 V ac is only 34 VA maximum, thereby making it possible to power the discrete synchronizing relays and the speed and voltage control relays from the bus vt, i.e., the power source is in parallel with the bus potential input terminals. Since the vt circuits are closed only when a synchronizing is required, the relays are themselves powered only when a synchronization is anticipated.

Physically, the new relays are semi-flush panel mount designs, also suitable for standard rack mounting. Each relay, the synchronizing relay (25A), shown in Figure 2 and the generator control relay (15/90) shown in Figure 3, occupies two standard rack units in height.

Both relays include accurately calibrated operator-selected dial settings to establish the desired control operation and front panel LED's to indicate to the operator that setpoints are within limits and reveal contact closure of the speed and voltage output circuits.

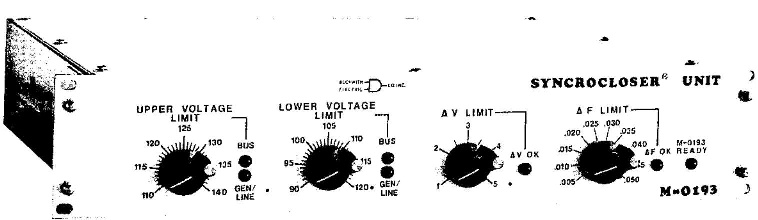


FIGURE 2 Synchronizing Relay M-0193 (25A)

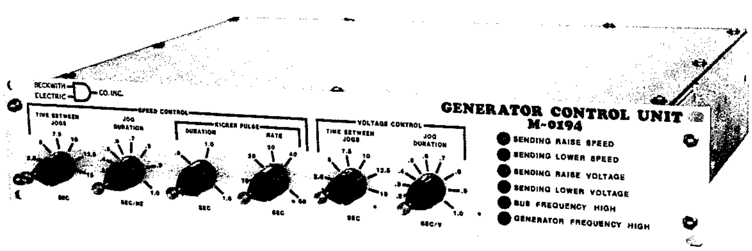


FIGURE 3 Speed and Voltage Control Relay M-0194 (15/90)

The new relays differ from the old in several major operational aspects:

1) recognition of variability of breaker closing time and the ability to configure the M-0193 control accordingly;

2) an accuracy of the closing angle of ±0.5° at slip frequencies and breaker closing times typical of the utility environment;

3) most notably, the application of proportional pulse width control for the speed jog output.

MOTOR SPEED CONTROL

The control of the turbine/generator speed, in order to attain synchronism with the system, may be accomplished by pulsing or jogging of a drive motor. The pulse train of speed jogs may take one of several forms.

The simplest form of control is that which employs a fixed pulse time with a fixed dead time. Using this approach, there is no feedback and no modification of the control action as synchronous speed is approached. Such a system must be calibrated to not cause overshoot and resulting hunting (under damped response) if the ratio of "on" to "off" period is large (Figure 4a), or an unacceptably slow attaining of synchronism (over damped response) if the ratio is small (Figure 4b).

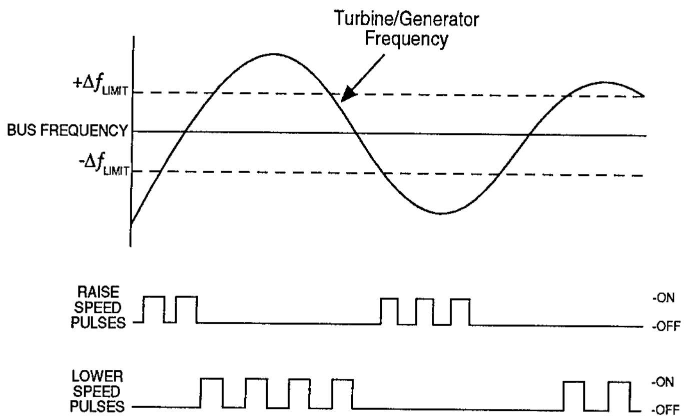


FIGURE 4a Fixed Pulse/Fixed Dead Time Control Action—Hunting Response

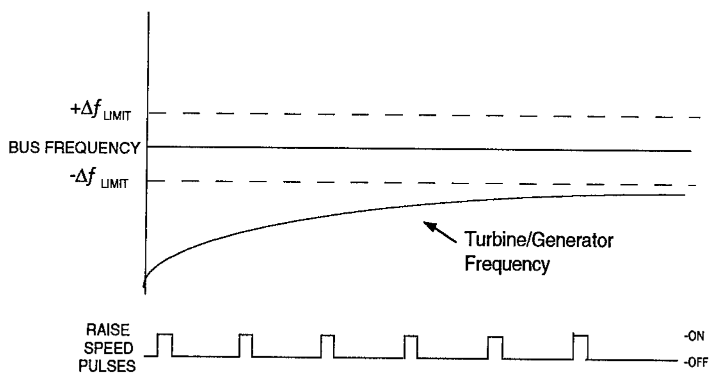


FIGURE 4b Fixed Pulse/Fixed Dead Time Control Action—Unduly Slow Reponse

Proportional control adds a degree of sophistication to the operation. With proportional control, the dead time or the pulse time is a function of the generator digression from synchronous speed. Thus, using feedback control, the ratio of the "on" pulses to "off" time is reduced as the speed closes in on the synchronous speed. This permits rapid attainment of the desired speed without the tendency for overshoot described earlier.

One form of proportional control uses a fixed pulse width with slip cycle dead time. This control action, while an improvement from the fixed pulse/fixed dead time response, suffers in systems which include a significant time lag between the jog pulses and the subsequent machine response.

Hydroelectric generator response times to a short speed jog may be 3 to more than 10 seconds so that while systems employing the fixed pulse/slip cycle dead time control may be calibrated to not cause overshoot, the resulting speed adjustment may remain unacceptably slow.

A more suitable form of proportional speed control, and that used in the change-out reported in this paper, involves proportional pulse width control with a fixed dead time, per Figure 5.

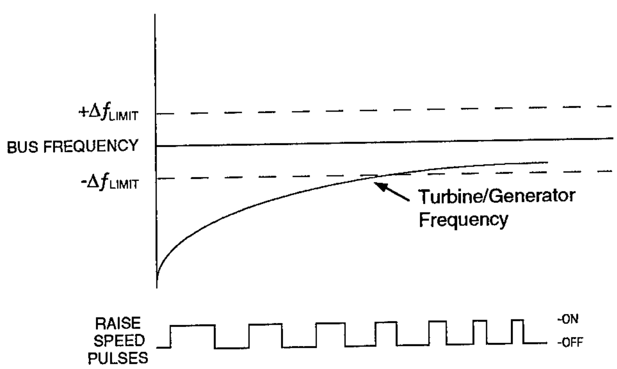


FIGURE 5 Proportional Pulse Width/Fixed Dead Time Control Action

In this proportional mode, rather than varying the rate at which a fixed time pulse is generated, the system accomplishes proportionality by varying the "on" time of a corrective pulse as a function of the degree of remaining speed mismatch. This mode is found to be preferable, especially for the high inertial load of a hydroelectric generator where the change of turbine speed is gradual and a longer "on" pulse is desired for a quicker response.

CONTROL PERFORMANCE

The strip chart recordings, Figures 6a and 6b, display the relative machine response to fixed pulse width/fixed dead time control and proportional pulse width/fixed dead time control at one system where the upgrade was made.

Figure 6a shows the response of the system to the old fixed pulse/fixed dead time control. For this case, the machine was operating at about 59.7 Hz when the control was enabled. It is apparent from the figure that the speed adjustment commanded by the synchronizer was too great, as the slip frequency is fluctuating wildly (a change of about 0.9 Hz in 20 seconds) resulting in an overshoot of the synchronous speed 5 times in 1.5 minutes. (Note: the chart recorder plots only the magnitude of the slip frequency.) This test was terminated as the slip frequency pattern was completing 3 cycles.

The new proportional pulse width/fixed dead band control system was enabled at about 59.5 Hz as illustrated in Figure 6b. In this figure, the slip frequency plot shows a gradually diminishing slip frequency between the generator and the bus. The slip frequency plot reveals that there is no overshoot experienced. Synchronism is attained and the circuit breaker is closed in 2.5 minutes.

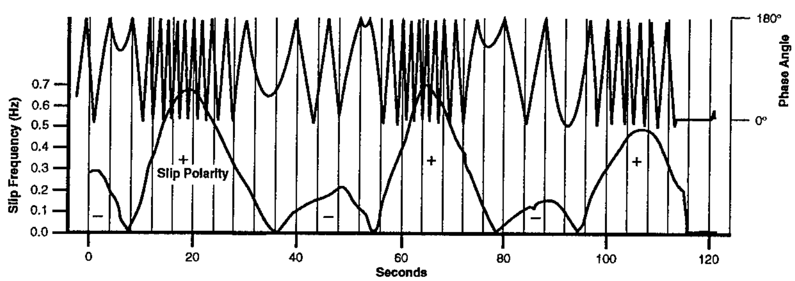


FIGURE 6a Machine Phase Angle and Slip Frequency vs. Time Using Fixed Pulse Width/Fixed Dead Time Control

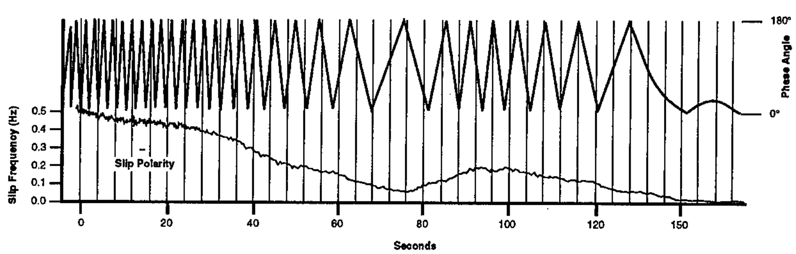


FIGURE 6b Machine Phase Angle and Slip Frequency vs. Time Using Proportional Pulse Width/Fixed Dead Time Control

CONTROL OPERATION

The control system is illustrated by the block diagrams of Figures 7 and 8 for the synchronizing relay (M-0193) and the speed and voltage control relay (M-0194) respectively.

Synchronizing Relay (Figure 7)

Each input (Gen/Line, Bus) is scaled down from 120 V ac to 6 V ac. The scaled-down voltage is converted into a dc voltage by the Converters. Each Converter is an active full-wave rectifier and filter which eliminates the usual highly temperature-dependent diode drop of conventional full-wave rectifiers.

Upper and Lower Voltage Comparators compare the output of each ac to dc Converter to a highly stable hybrid 10 V reference. LEDs located on the front panel indicate the condition of each input (Gen/Line, Bus) with reference to upper and lower voltage limits.

A circuit provides a voltage which is directly proportional to the absolute value of the difference between two or more voltages. This circuit, the Absolute Value Difference Amplifier, is used to precisely evaluate the difference voltage ΔV between the two inputs. A Comparator compares this difference voltage to a portion of the highly stable 10 V reference. Again, a front panel LED indicates the status of ΔV.

Two Zero-Crossing Detectors generate rectangular waveforms at zero-crossings of each filtered input. The AND-gate provides a pulse width proportional to the phase angle between each input. A seven-pole Active Filter provides the required averaging without the excessive time delay of conventional RC filters. Phase angle voltage φ varies from 0 to 10.0 V for a 0 to 180° phase change, and back to 0 V for an additional 180° phase change. The phase angle voltage slope $d\varphi/dt$ is directly proportional to the difference frequency ΔF. This derivative is bipolar, depending upon the direction of phase change. By taking the absolute value of $d\varphi/dt$, continuous monitoring of ΔF is achieved without the usual "dead band." Absolute value and time derivative $d\varphi/dt$ are functions of the Absolute Value Amplifier and the Precision Derivative Amplifier. This is followed by a ΔF Comparator with a front panel LED indicator, which indicates in- or out- of band conditions.

Breaker closing time is generated by selecting one of a series of programmable times, ranging from 20 to 200 ms. The times are a function of the Breaker Closing Time Network. A Multiplier, which is programmable from X1 to X4, allows a full range of times to 800 ms. The operation of breaker closing time is accomplished by dividing and scaling the magnitude of $d\varphi/dt$ and comparing this to φ. When the comparator of φ and $|d\varphi/dt|$ senses that they are equal, a breaker-close pulse is provided. The comparator is arranged to provide a pulse only when φ is approaching zero degrees, which eliminates the possibility of closing "going away from zero."

All circuit parameters are continuously monitored by the logic circuits. This includes the Power Supply voltages, all input voltages and frequency conditions, and power relay charging capacitor voltage. The output power relay can only be closed when sufficient charge is available in the capacitor. Charge is allowed to "flow in" when all input voltages and frequency conditions are met. This scheme virtually eliminates any possibility of false closing under any conditions.

Signals required by the speed and voltage control relay are developed in the synchronizing relay and transferred to that relay by an interconnecting cable.

Speed and Voltage Control Relay (Figure 8)

The M-0194 control works in conjunction with the M-0193 unit, with all sensing signals connected through the connector cable. The only rear terminal connections required to the M-0194 are the line power supply and a jumper to inhibit relay closure.

The phase angle between the bus and generator inputs is precisely measured by the Phase Angle Detector circuit which compares two square waves from the M-0193 voltage input secondaries. This circuit provides a pulse width proportional to the phase angle between two inputs and a dc voltage proportional to the phase angle.

A Voltage Level Comparator monitors the voltage level of the bus and generator sources to ensure that the Phase Angle Detector can precisely measure the phase difference. If one of the two source voltage levels drops below 25% of nominal, this level detector will inhibit the Phase Angle Detector circuit.

A Derivative Amplifier measures the slope of the phase angle and provides a triangular wave. This slope is proportional to the difference in frequency (ΔF).

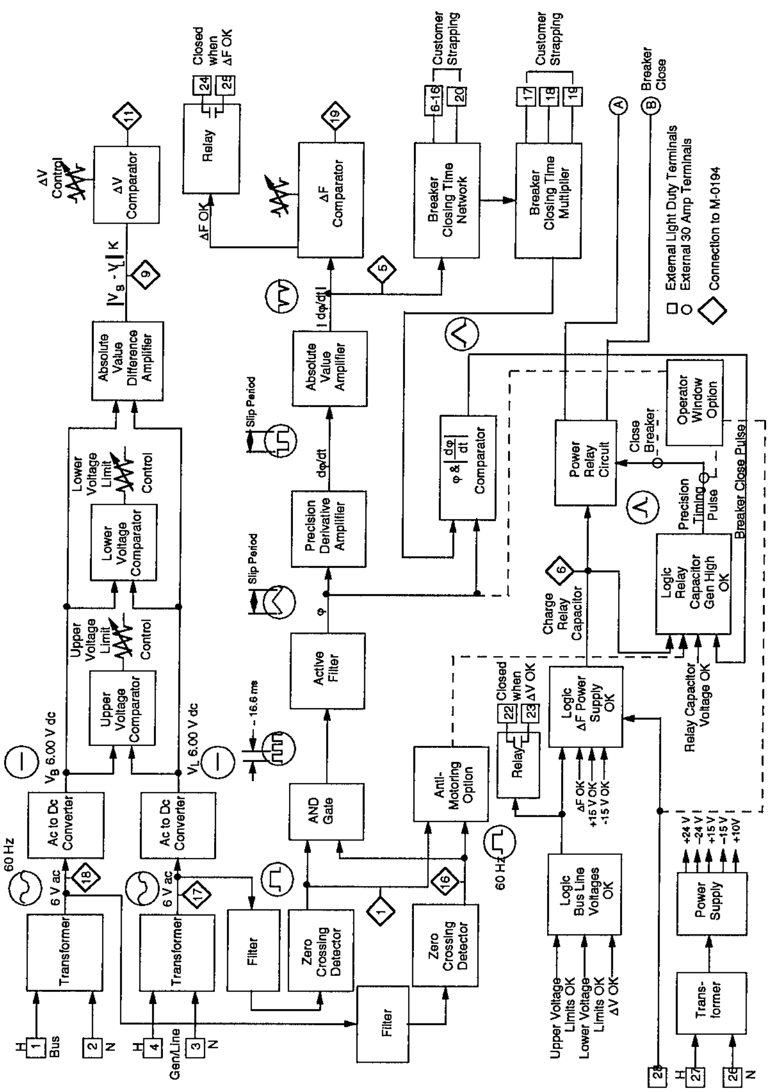


FIGURE 7 Block Diagram of Synchronizing Relay, M-0193

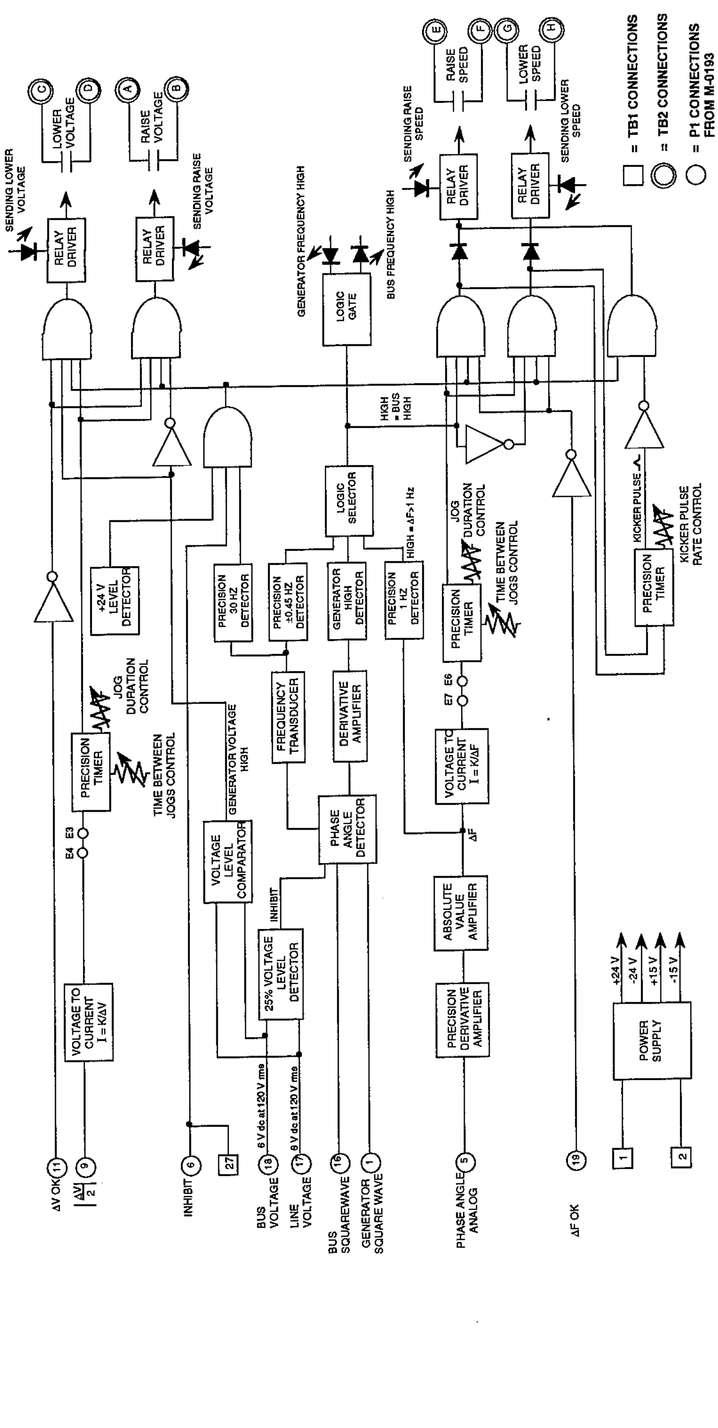


FIGURE 8 Block Diagram of Speed and Voltage Control Relay, M-0194

The highly accurate frequency detector (Generator High Detector) will detect whether the generator's frequency is higher or lower than nominal. This signal will be one of the three inputs to a Logic Selector to give correct indication and commands for speed matching.

The Frequency Transducer precisely senses the generator frequency and produces a voltage level which is proportional to the frequency. Another Precision Detector is used to detect whether the generator's frequency is higher than 60.45 Hz or less than 59.55 Hz, presuming a nominal 60 Hz system. This signal is another input to the Logic Selector for indication. The voltage level from the transducer is also detected and will inhibit all of the command outputs if the generator frequency is less than 30 Hz. Speed control is inhibited for generator frequencies below 30 Hz.

The Precision Derivative Amplifier measures the rate of change of the Phase Angle Analog Voltage from the M-0193. This signal is then rectified in the Absolute Value Amplifier. The output of the Absolute Value Amplifier is a voltage, proportional to the magnitude of ΔF between the generator and the bus voltages.

A comparator circuit compares the magnitude of ΔF from the Derivative Amplifier to a fixed reference voltage. The output of the Precision 1 Hz Dectector goes "high" when the magnitude of ΔF is greater than 1 Hz. This signal is used by the Logic Selector to choose between the Precision ±0.45 Hz Detector or the Generator High Detector. When the magnitude of ΔF is greater than 1 Hz, the Precision ±0.45 Hz Detector circuit indication is used. When the magnitude of ΔF is less than 1 Hz, the Generator High Detector is used. The Generator High Detector is not reliable above 1.5 Hz frequency difference.

The output of the Voltage to Current Converter is inversely proportional to ΔF and is converted into a current source for the input of the Precision Timer. When the timer times out, pulses are provided for closing either the Raise Speed or the Lower Speed relay, with the pulse width directly proportional to ΔF when proportional control has been selected.

Similar to the proportional speed circuit, a signal inversely proportional to ΔV is derived and is converted into a current source for the input to a second Precision Timer. The pulse width from the output of the timer is directly proportional to the actual voltage difference.

A Voltage Level Comparator compares two dc voltage levels which are stepped down from the bus and generator voltage sources and rectified by the M-0193 input circuit. This amplifier determines the higher voltage level of the two and will enable the proper relay for either a Raise or Lower Voltage command.

A charge current from the speed change output goes to the third Precision Timer. This generates a short pulse after a time delay as set by the Kicker Pulse Rate dial on the M-0194. This pulse only exists when the ΔF is within the limit set on the M-0193. If the speed matching circuit sends either a Raise Speed or a Lower Speed Jog, this timer is reset. An adjustable raise speed jog is sent when the kicker pulse timer times out. The kicker pulse length is set by a front panel dial, marked Kicker Pulse Duration, and can be adjusted from 0.1 to 1.5 sec.

CONCLUSIONS

The upgrading of the system reported enhances the performance of the plant automatic synchronizing scheme. The units involved will start remotely, making it imperative that the synchronizing function be dependable.

The proportional pulse width with a fixed dead time method of control from speed matching has been demonstrated and confirmed by oscillographs to provide an excellent method of control for the very high inertial conditions found in hydroelectric generating stations.

Replacement of the Wood Stave Penstock and Turbine Runners at Tuxedo Hydro Plant

William A. Maynard[1]

Abstract

Duke Power Company replaced the 1920's vintage wood stave penstock and turbine runners at its two unit, 5 megawatt Tuxedo Hydro plant near Hendersonville, North Carolina. The penstock replacement was necessary due to a massive earth slide that caused the washout and rupture of a 75 meter long section of penstock in 1987. The runner replacement was necessary due to the age and condition of the original runners, as well as a desire to maximize energy output with modern, more efficient runners.

This paper presents a discussion of the design, fabrication, construction, and testing of the new penstock and runners. The design presented unique challenges due to cost constraints, the poor penstock foundation, the major slide area and several minor slide areas, considerations regarding normal wood stave penstock leakage, and the vagaries of integrating new components with the balance of the 70 year old plant. Construction considerations included removal of the original penstock, penstock foundation preparation, difficult access and working conditions due to the rugged terrain, and the constructor's lack of experience constructing wood stave pipe.

The plant began generating electricity again in December of 1991 following successful testing of the penstock and runners.

Introduction

Tuxedo Hydro Plant is located in Henderson County, North Carolina on the Green River between Hendersonville and Saluda, North

[1]Engineer, P.E., Duke Power Company, P. O. Box 1006, Charlotte, NC 28201-1006, (704) 382-5671

Carolina. The plant was built by the Blue Ridge Power Company and placed into operation in 1920. Duke Power Company (DPC) acquired the plant in 1927.

Plant structures include a 36.9 meter high by 74.4 meter long single arch concrete dam, penstock and surge tank, and powerhouse. The powerhouse contains two 2500 kilowatt rated capacity vertical units operating under a 87.5 meter (287 foot) head. The powerhouse is located about 1.5 kilometers downstream of the dam.

The penstock originally consisted of a single 2.44 meter diameter wood stave penstock (hereafter referred to as "8 ft. penstock") extending 1439 meters (4720 ft.) from the dam to the surge tank, and two parallel 1.52 meter diameter by 183 meter (600 ft.) long wood stave penstocks (hereafter referred to as "5 ft. penstocks") from the surge tank to the powerhouse. The 8 ft. penstock remained essentially level along its entire length, with a 16 meter (23 psi) static head. From the surge tank, the 5 ft. penstocks drop 69.5 meters (228 ft.) to the powerhouse at a gradient of approximately 37.5%. The two 5 ft. penstocks are subjected to a 85.6 meter (122 psi) static head at the powerhouse. The 7.6 meter diameter by 28 meter high cylindrical surge tank is constructed of reinforced concrete.

The original penstocks were constructed of 3 1/2" thick by 5 1/2" wide Cypress staves girded by 3/4" diameter steel bands at varying spacing (150 mm spacing at the dam and 50 mm spacing at the powerhouse). The 8 ft. penstock was constructed on an excavated bench along the side of a steeply sloping mountain, with embankment fill placed in several ravines along the alignment. This bench runs roughly parallel to and above the old river bed between the dam and the surge tank. The penstocks were supported on cast in place concrete cradles. Each 5 ft. penstock was restrained from vertical movement under surge by two concrete anchor blocks. The penstocks were connected to the intake, surge tank, and powerhouse by attachment to steel thimbles. One end of each thimble fit inside the wood stave pipe and the other end embedded in concrete.

The penstocks performed well over the years primarily due to the unique ability of wood stave pipe to tolerate large amounts of foundation movement. The cast in place cradles had been placed directly on the unimproved subgrade, resulting in differential settlement throughout the pipeline. Additionally, slide induced horizontal movement occurred at several locations over the years. One section of 8 ft. penstock was able to withstand a horizontal movement of 4.5 meters in 1964. The penstock did not rupture and was left in its deflected position with 6 new cradles added for support.

A major landslide occurred in 1936, collapsing a 90 meter length of 8 ft. penstock. A man-made embankment fill and retaining wall that had been placed to support this section of penstock across a ravine slid off the side of the mountain and into the river after several days of heavy rain. Following this failure a steel trestle on concrete piers was built over the washed out area, and a new section of penstock constructed on the trestle. This section was rebuilt with untreated Douglas Fir wood staves.

The late 1970's and early 1980's saw an increase in the amount and severity of leaks in the aging 8 ft. penstock. Site inspections and stave testing revealed that the pipe was nearing the end of its service life due to wood stave deterioration. The 5 ft. penstocks were in somewhat better condition having been completely replaced in 1946 with untreated Douglas Fir wood stave pipe. The concrete cradles, steel thimbles, and steel bands were in good condition.

Penstock Failure

On the morning of June 10, 1987, another massive earth slide occurred in a man-made embankment fill supporting the penstock. The embankment fill and a 75 meter long section of 8 ft. diameter penstock collapsed, washing the fill material, penstock, and vegetation downslope and into the Green River. Buried in this embankment fill was a 42 in. concrete box culvert which carried a small stream through the fill, and a 12 in. diameter water main supplying the town of Saluda. Parts of both the culvert and water line were also washed out.

An engineering evaluation of the situation was performed on the day after the failure by DPC and Law Engineering. Of particular concern was the stability of the state highway bridge which is located about 10 meters from the edge of the failed area. It was quickly determined that the bridge is founded on rock so there was no danger of undermining the bridge.

The evaluation determined that the failure was likely precipitated by a shear failure of the steep embankment fill slope facing the river, with the failure surface extending back under the penstock. Thus undermined, the penstock ruptured. The force and flow of water from the ruptured penstock washed away most of the original fill, and a 15 meter deep gorge was created. The initial embankment failure was probably initiated by saturation caused by either penstock leakage or leakage from the buried water main.

Fortunately the headgate was closed when the failure occurred, limiting the extent of damage to that caused by the volume of water in the

penstock and surge tank, and potentially preventing the drainage of up to 11 million cubic meters of water in the reservoir.

Within days of the failure, a dumped rock weir was constructed in the old river bed just upstream of the powerhouse in order to create a temporary settling pond for mud and debris from the washout. This settling pond was cleaned out many times. The exposed slopes of the failed area were stabilized by grading, sowing grass, and placing riprap.

1987 Penstock Failure

Post Failure Plant Disposition

With the mud contained and the failed area stabilized, the disposition of the plant was addressed. The following alternatives were identified:

A. Retire the plant; maintaining the dam and grounds.

B. Lease or sell the plant.

C. Rehabilitate and modernize the plant. Replace only the washed out section of penstock, and line the interior of the rest of the 8 ft. diameter penstock with waterproof membrane. The two 5 ft. penstocks would

remain without modification. Install new, more efficient turbine runners.

D. Replace the entire penstock including 5 ft. lines, rework and stabilize the penstock bed (foundation), and install new more efficient turbine runners.

A technical and economic evaluation of these alternatives led to the decision to replace the entire penstock and rehabilitate the plant for an additional 40 years of service.

The scope of replacement of the penstock included removal of all concrete cradles from the dam to the surge tank, excavation and stabilization of several slide or potential slide areas along the 8 ft. penstock bed, construction of a new steel trestle over the failed area, painting and repair of the old trestle at the 1936 slide location, painting and reuse of existing thimbles, and construction of a new wood stave penstock along the same alignment as the old penstock. The existing concrete cradles supporting the 5 ft. penstocks would be repaired as necessary and reused.

Although consideration was given to rebuilding the penstock with steel pipe, it was advantageous to stick to the original concept of wood stave construction. The advantages are simple but important:

1. The original penstock provided 67 years of low maintenance service.

2. Wood stave pipe is uniquely suited for construction in rugged terrain where access is limited.

3. The demonstrated ability of wood stave pipe to accommodate quite large foundation movements allows construction on a much lower quality subgrade than would be necessary for a steel pipeline.

4. The method of construction is proven. There are many wood stave pipelines in service worldwide.

Design and Construction

DPC project engineers performed all foundation, concrete, and structural steel design. The wood stave pipe vendor, Canbar, Inc., of Ontario, Canada, designed the penstocks and wood cradles. An analysis by DPC engineers determined that the 8 ft. penstock could be replaced

with a single 2.13 meter (7 ft.) diameter penstock. This reduction in penstock diameter was crucial in cost justifying the project.

The new penstock is constructed of creosote treated Douglas Fir tongue and groove wood staves girded by 3/4 inch diameter steel bands. The 7 ft. penstock is supported by creosoted heavy timber cradles on 125 mm (5 inch) thick precast concrete pads.

All construction work was performed by DPC construction crews. Although DPC employs experienced power plant construction and maintenance crews, there was no experience in constructing wood stave pipe. There was some apprehension going into the job as to whether the construction schedule could be met.

The old penstocks were expeditiously removed by cutting into short full round lengths with chain saws, then a small crane was used to place these segments on trucks for removal from the site. Following this, the old concrete cradles were removed, the entire penstock bed regraded, and a layer of compacted crusher run stone placed. The old cradles were placed at several locations along the downhill slope adjacent to the penstock bench. This application saved the cost of disposal of the cradles, as well as the cost of riprap that otherwise would have been necessary for bed stabilization. Unstable rock at several locations along the uphill side of the penstock bench was pinned with 3 to 6 meter long resin grouted Dywidag rock bolts.

Three locations with a history of significant bed movement were stabilized. Just downstream of the dam, a 1 to 2 meter depth of loose overburden was excavated to rock, and a 43 meter long retaining wall dowelled into the rock. Compacted crusher run backfill with washed stone for drainage was placed behind the wall. The other two locations were excavated to competent material, then backfilled with a reinforced earth fill. Tensar geogrid and compacted fill were placed in 450 mm to 600 mm layers, with a layer of compacted crusher run at the top. Several wet areas and small springs uncovered in these excavations were treated with a geotextile drainage composite and french drains prior to backfilling.

The failed area was treated by installing corrugated metal pipe in place of the washed out box culvert, regrading, and erecting a new steel trestle. Twelve inch diameter pipe piles were driven and filled with concrete, and four structural steel bents were erected to support two W36 steel beams. Bolted to these beams are structural steel cradles spaced 2.1 meters apart. All structural steel is A588 Weathering Steel, painted only at the "wet face" in contact with the wood penstock.

Construction access to the 7 ft. penstock was by means of improving old construction roads at three locations; between the dam and the new trestle, the new trestle and the old trestle, and the old trestle and the surge tank. The penstock was built from the most remote points (dam and surge tank) towards the middle access point such that materials were delivered utilizing the crusher run surfaced penstock bed for access to the work points.

Access to the steeper 5 ft. penstocks lines was much more difficult. A construction road was built from the top of the hill at the surge tank down to staging platforms at the top and at the one-third points on the hill. A crane was set up at these points to move materials to the work point. These penstocks were built from the bottom of the hill up.

Upon removal of the old penstock, the embedded steel thimbles were found to be quite out-of-round. Since it was anticipated that a watertight connection of the wood pipe to these out-of-round thimbles could not be achieved, a new steel "transition piece" was quickly fabricated for each of the four 5 ft. diameter thimbles. Each thimble was temporarily rounded out with spiders and a full penetration weld to the new transition piece was made. The penstock staves were then mated to the transition pieces. Steel reducers were similarly attached to the out-of-round 8 ft. diameter thimbles.

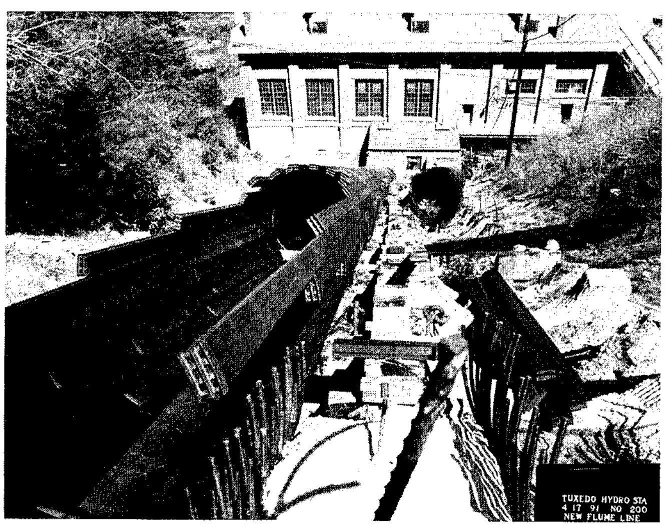
The first penstock staves were placed in March of 1991. After a short length of 7 ft. penstock was constructed, work began on the 5 ft. penstocks at the powerhouse. All staves were hand carried to the point of construction, set into place on the form, and driven up to the end-adjacent stave with sledge hammers. It is interesting to note that the method of construction in 1991 was essentially the same as that used on the original penstock in 1920.

Efficient and safe construction of the 5 ft. penstocks was a challenge due to steep terrain, inexperienced construction crews, unpleasant working conditions, and poisonous snakes. The heavily creosoted wood staves required the wearing of coveralls and rubber gloves in up to 95 degree F temperatures. Slow going and a behind schedule completion of the 5 ft. lines was the result. Work was completed without a serious injury, however. Following completion, the two 5 ft. penstocks were pumped full of water to prevent drying and shrinkage over the summer while the 7 ft. penstock was being constructed.

Construction of the 7 ft. penstock line went smoother due to the level terrain, easier access, and the now experienced construction crews. Under the guidance of a Canbar construction advisor, two crews were utilized building toward each other; one from the surge tank and one from

the dam. The early stages of construction of the 7 ft. penstock revealed that building the curves was much more difficult and time consuming than anticipated. Because of this, project engineers decreased the radius of the curve just upstream of the surge tank (one of the more severe curves) by quickly fabricating a 13 degree mitered steel extension that was welded to the surge tank reducer. The result of this design change was an estimated schedule and labor savings of one week.

Despite a late start and the initial problems building the curves, the 7 ft. diameter penstock was built much quicker than anticipated, resulting in completion of all penstock construction over one month early.



5 Ft. Penstocks Under Construction

Testing and Operation

The headgate was cracked open to begin filling on November 7, 1991. Filling took place over the next several days while installation of the new turbine runners was being completed.

With the penstock completely full on November 12, and the pressure rising towards full static head, a noticeable increase in leakage between the wood staves occurred. Several locations experienced washing of the bed, such that an emergency use of previously stockpiled sand bags became necessary. While no permanent bed damage occurred, around-the-clock monitoring of the penstock was initiated, and

plastic sheeting was temporarily placed on the bed at locations of heavy leakage such that additional washing was prevented. Additionally, concrete was placed under an approximately 150 meter length of the penstock as a permanent means of directing leakage water away from vulnerable parts of the bed.

Following pressurization to full static head, the leakage rate began declining as the wood staves swelled and sealed. By mid-December, leakage had declined sufficiently that the around-the-clock monitoring was discontinued and most of the plastic sheeting removed.

Testing of the rebuilt turbines and the new penstock was begun on December 12, 1991, and on that day the plant generated electricity for the first time in 4 1/2 years. Unit load rejection testing found the new penstocks to be significantly more stable and watertight than the old penstocks had been.

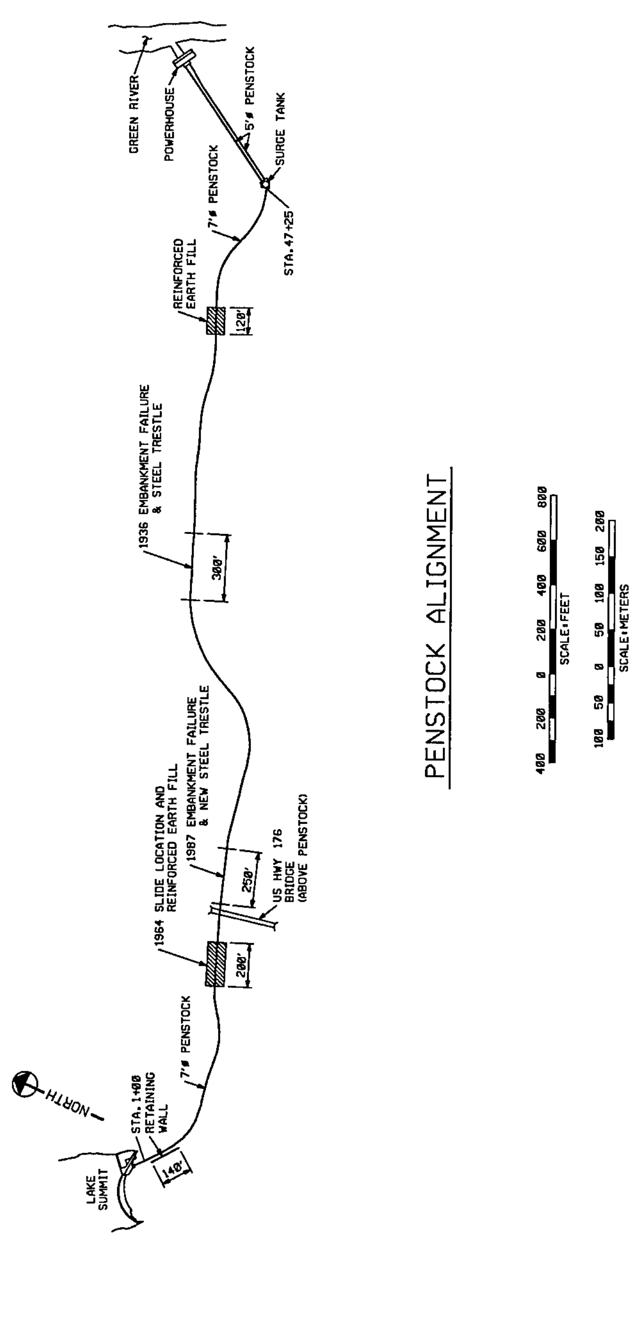
The refurbished plant can generate 33% more power than before, each unit capable of generating 4000 kilowatts. Water samples taken before and during testing confirmed that the creosoted penstock was not releasing detectable amounts of creosote into the river.

The penstock leakage rate continued to decrease until late summer of 1992 when it appeared to stabilize. At that time, a crew traversed the entire 7 ft. penstock plugging the remaining leaks with wood plugs and wedges and "candlewick" twine. While the pipe is not completely watertight, the leakage is well within acceptable rates for wood stave pipe. The penstock bed has performed well, with only minor sloughing at a few locations and no noticeable settlement.

Conclusion

Plant testing verified the integrity of the new wood stave penstock as well as the increased efficiency of the new turbine runners. While some problems were encountered due to unexpectedly heavy initial penstock leakage at some locations, the project was successfully completed.

Meeting project schedules and budgets was of some concern going into the job due to the limited knowledge and lack of experience regarding wood stave pipe construction by DPC engineers and constructors. The project was completed on schedule and well under budget however, primarily due to cost cutting efforts during design and construction, as well as teamwork and flexibility by DPC project engineers and constructors. The plant rehabilitation was successfully completed at a cost of $730 per kilowatt, with an expected life of over 40 years.



PENSTOCK ALIGNMENT

"RE-POWERING NINE MILE HYDROELECTRIC PROJECT"

Joseph A. Kurrus, Member[1]
Rick Zilar[2]
Edward Schlect[3]
Reynold A. Hokenson, Member[4]
Jim H. Rutherford, Member[5]

ABSTRACT

Washington Water Power (WWP) is pursuing rehabilitation of its hydroelectric systems. This is being done to replace aging equipment, and to maximize energy production by increasing efficiency and capacity. Using a variety of design and construction techniques, WWP will modernize and upgrade it's Nine Mile Hydroelectric Development. The result will extend the life of the plant 50 years, increase generating capacity by 110% and provide reliable operation.

INTRODUCTION

Nine Mile Hydroelectric Development (HED) was completed in 1910 by the Spokane and Inland Empire Railway. Their intent was to provide for increased industrial growth in the area, including an electrified railroad. The project was purchased by WWP in 1925. It consists of a powerhouse structure, spillway, and integral concrete gravity left abutment. Ten operators cottages were completed in 1930 but are no longer used. The powerhouse structure and cottages are listed on the National Register of Historic Places.

The Nine Mile HED is part of the Spokane River system which consists of five individual hydroelectric developments (FERC License No. 2545). It is located at River Mile 58 on the Spokane River, upstream of and tributary to the Columbia River in Northeastern Washington State. The Nine Mile pool covers 141 hectares at low flow and extends about 11.1 kilometers

[1]The Washington Water Power Company, P.O. Box 3727, Spokane, WA 99220
[2] The Washington Water Power Company, P.O. Box 3727, Spokane, WA 99220
[3] The Washington Water Power Company, P.O. Box 3727, Spokane, WA 99220
[4]R. W. Beck and Assoc., Fourth and Blanchard Bldg., 2101 Fourth Avenue, Seattle, WA 98121
[5]R. W. Beck and Assoc., Fourth and Blanchard Bldg., 2101 Fourth Avenue, Seattle, WA 98121

upstream. This HED is operated as a Run of River plant, that is river flows are generally matched through the plant as practically as possible. When river flows exceed the plant capacity (147 cms), water is allowed to spill over the spillway. The average River flow is about 204 cms at Nine Mile HED.

The company has chosen to replace two of the four horizontal turbine/ generators. New controls, substation and related equipment will also be upgraded. In addition WWP is replacing the existing bridge cranes and intake gates as part of its maintenance efforts. A pool raise of 1.5 m by installing a rubber dam is also being considered.

A non-capacity license amendment and necessary permits are being pursued for the rehabilitation project. The proposed pool raise project will also require a license amendment, if it is decided to proceed. The replacement of the intake gates is considered a maintenance project and minimal permits are required.

EXISTING FACILITIES

SPILLWAY

The spillway structure is an ogee type approximately 17.7 m high by 68.6 m long. Wooden flashboards are installed on top of the spillway to raise the normal pool 3 meters. During high flows these flashboards are removed.

POWERHOUSE

The Powerhouse structure is located in the river and comprises the part of the dam. It contains four bays, one for each of the horizontal Francis turbines with quad runners. The generators are located adjacent to the turbines on the lower (generator) floor. Each is connected to the turbine by a horizontal shaft and separated by a bulkhead. The small control room is positioned above and between generator units #2 and #3. A bridge crane is located above the generators and runs the length of the lower floor. (Fig. 1)

The middle floor of the powerhouse houses the switchgear, station service, UPS and other ancillary systems. Originally, it housed four GSU transformers as part of an interior substation. This floor is directly above the generator floor. The upper (access) floor is upstream and above the switchgear floor. The upper floor originally housed the circuit breakers and bus which connected directly to the distribution system. Outside access is available through two large doors. A bridge crane is positioned to run the entire length of this floor as well.

LEFT ABUTMENT

The left abutment structure provides the final closure of the dam to the left (looking downstream) of the powerhouse and spillway. It is a solid concrete gravity structure.

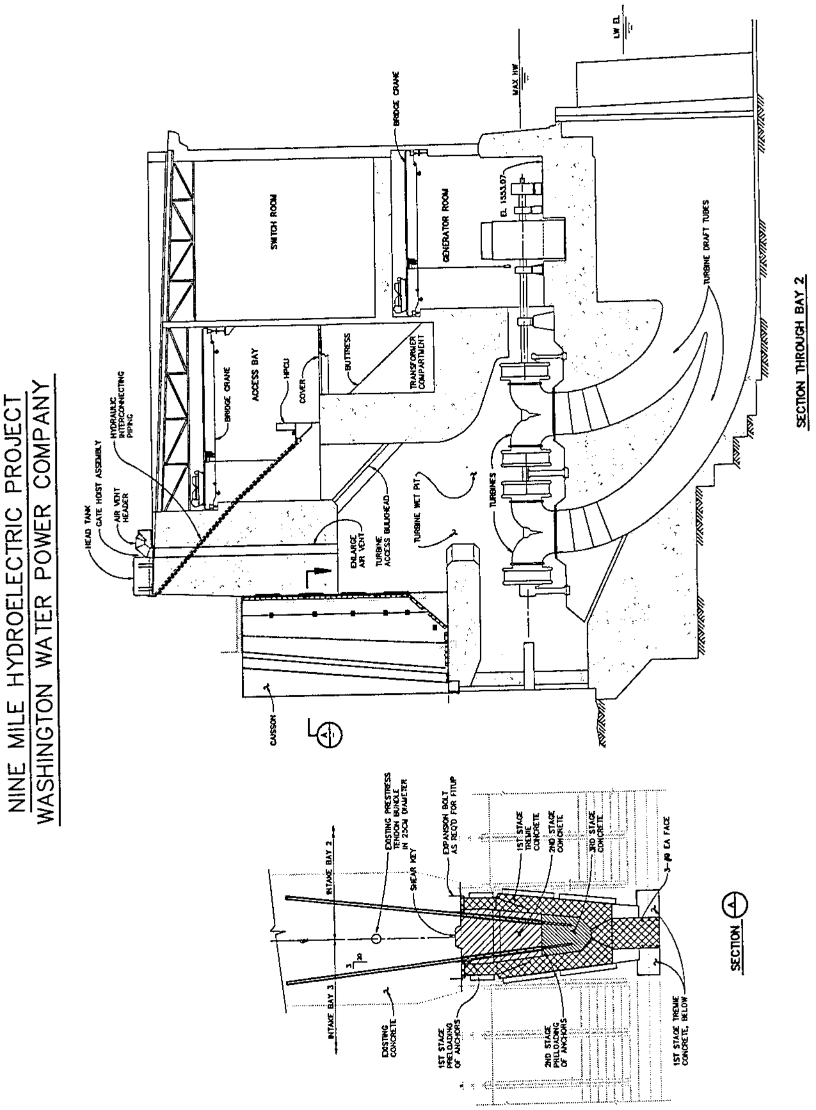

NINE MILE HYDROELECTRIC PROJECT
WASHINGTON WATER POWER COMPANY
HEAD TANK
GATE HOIST ASSEMBLY
AIR VENT HEADER
HYDRAULIC INTERCONNECTING PIPING
BRIDGE CRANE
ACCESS BAY
HPCU
COVER
BUTTRESS
TRANSFORMER COMPARTMENT
SWITCH ROOM
GENERATOR ROOM
EL 1553.07
BRIDGE CRANE
MAX HW
LW EL
TURBINE DRAFT TUBES
TURBINES
TURBINE WET PIT
TURBINE ACCESS BULKHEAD
ENLARGE AIR VENT
CAISSON
SECTION THROUGH BAY 2
INTAKE BAY 2
INTAKE BAY 3
EXISTING PRESTRESS TENDON BUNDLE IN 23CM DIAMETER
SHEAR KEY
EXPANSION BOLT AS REQ'D FOR FITUP
1ST STAGE TREMIE CONCRETE
2ND STAGE CONCRETE
3RD STAGE CONCRETE
EXISTING CONCRETE
1ST STAGE PRELOADING OF ANCHORS
2ND STAGE PRELOADING OF ANCHORS
1ST STAGE TREMIE CONCRETE, BELOW
SECTION A

FIGURE 1

SUBSTATION

The substation consists of an outside generator step-up transformer (GSU) next to the powerhouse and three circuit breakers. The generators operate at 2.3 kV and provide about 3.4 MW each. The GSU provides 60 kV for transmission.

EVALUATION PROCESS

ALTERNATIVES

Over 200 different alternatives for the Nine Mile HED Redevelopment project were evaluated. Alternatives for the replacement of two, three, or four units within the existing powerhouse and a new powerhouse were considered. Several different pool raise scenarios were also investigated. A matrix was then created by combining all the alternatives.

EVALUATION

Cost estimates and energy production estimates were produced for each alternative, including the base case. The base case is defined as the existing facilities with the improvements necessary to extend the service life equal to that of the other alternatives.

The electrical energy production of each alternative was estimated by modeling historical river flows.

Discounted cash flow modeling was used as the economic screening criteria. All cash flows were projected out beginning with the start of construction through 35 years of on-line generation. These cash flows were then reduced by their resultant tax effect and discounted back to project start date using the corporate discount rate. The methodology employed compared incremental financial performance differences relative to the base case.

Net Present Value (NPV) - The total of the after-tax cash flows discounted back to present day dollars using the Company's weighted-average cost of capital.

Internal Rate of Return (IRR) - The discount rate that will bring the incremental cash flows back to a $0 net present value.

Profitability Index (PI) - A benefit-cost ratio calculated by dividing the present value of benefits by the present value of the capital costs.

IRR and PI are useful indicators of economic merit in situations where there are more economically viable projects than capital dollars to fund the projects. These two indicators measure performance relative to the capital investment whereas net present value will identify overall profitability.

Economic Sensitivity Analysis - Several sensitivities were tested against the alternatives. These included power values, income tax rates, discount rates, and capital expenditures.

Revenue Requirements Analysis - Used to measure the effect on the rates of the different alternatives using the revenue requirement model. Capital expenditures, although only capitalized in the year of completion, were reflected as negative cash flows in the year of occurrence. All cash flows were adjusted to reflect the effect of federal income tax rate.

SELECTED ALTERNATIVE

Rehabilitation Project - The economic evaluation led to the selection of the alternative of replacing two units. This alternative equaled the highest NPV of all alternatives and had the highest IRR and was the lowest capital investment. It involves replacing turbine / generator units 3 and 4 entirely, including controls, governors, exciters and voltage regulators. The new capacity for each unit is approximately 10 MW, which compares to about 3.4 MW for the existing units. This additional capacity is available primarily due to the increased hydraulic capacity and efficiency of the new units.

Rubber Spillgate, New Generators (pending pool raise) - Should the company decide to raise the pool at Nine Mile HED a rubber spillgate would be installed to replace the existing flashboards. Under most conditions the spillgate would automatically keep the pool level the same by adjusting the height of the spillgate. To take advantage of this extra head, the generators for Units 1 and 2 would be replaced.

PROJECT IMPROVEMENTS

NEW INTAKE FACILITIES

Trashracks - As part of the planning process for installation of new turbine-generator units, the structural condition of the existing intake gates and trashrack support structure was first investigated. Divers took video pictures that clearly showed that extensive corrosion of the support structures' principal structural members had occurred. Because the existing operating pool elevation is seasonally raised as much as 1.5 m during high flows, additional support structure and trashracks will be supplied. The trashracks and support structure are designed to support 3 m of differential hydrostatic head that was assumed to simulate a conservative, partial plugging of the trashracks.

Intake Gates - New intake gates will be installed to replace the existing 16 slide gates (4 per bay) with 4 fixed wheel type gates. One existing gate per bay is fitted with a sluice gate to allow filling of the turbine wet pits to equalize the hydrostatic load prior to raising the gates. They consist of steel frames and skin plates with supporting wooden seal members. Currently, extensive caulking and sealing of the existing gates by divers is required prior to allowing access to the turbine wet pits.

Based on cost, operation and maintenance considerations, it was determined that 4 new, fixed wheel gates powered by retracting hydraulic cylinders would be the best choice (1 per bay).

The new gates will be an assembly of structural steel weldments, consisting of horizontal plate girders integral with a skin plate on the downstream side of the frame, a bottom-sloping plate slab to improve flow, and vertical side plates. The gates are 4.6 m wide x 6.1 m high and designed for a head of 12.2 m and to close under 56 cms flow, automatically. Each gate weighs 31.7 metric tons.

To allow the gates to be installed, two sets of bulkheads consisting of welded structural steel plates and shapes to form water-tight compartments will be provided. Skin plates and internal girder webs and end plates will be reinforced with stiffener elements. The bulkhead elements will be bolted together into single three-section hinged assemblies and floated horizontally into position to provide water tight barriers for the intake. Bulkheads will be furnished with seals, pipes and valves to allow flooding and dewatering. Bulkheads will be 7.7 m wide by 8.8 m high and weigh 9.1 metric tons each. Two of the bulkhead sections will be fitted with a second set of seals to allow them to be used as a draft tube bulkhead.

Hoist - The gates are currently raised and lowered using a rack and pinion hoist powered by a moveable, electric power driven hoist car. If a unit goes to runaway conditions with the existing hoists, the time required to lower all four gates would probably exceed the safe runaway time of a unit. (Current industry practice requires automatic closure of an intake gate triggered by excessive velocity, rpm, or frequency monitors to protect a unit from sustained runaway). The existing hoist will be replaced with four hoists so each gate will open individually and quickly.

Intake Pier and Wing Wall Extensions - To allow the new intake gates to be installed and provide for an access bridge and service platform for the intake, three piers and the end walls of the existing intake are to be extended and raised. The first pier to be extended and raised was the pier between Units #2 and #3.

This extension involved the construction of a double-walled, steel caisson with the outer space between the caisson plates forming the outer wall, or water face, of the new pier. The dimensions of the caisson are 2.4 m wide by 5.5 m long by 13.4 m high. Shear lugs were welded to the 7.9 mm skin plate to engage concrete infilling, and internal cross braces, consisting of W sections spanning between the double walls, were used to prevent collapse during dewatering.

Before installation, the concrete pier nose geometry was checked with a template made with the assistance of divers to insure that a near water-tight fit would be achieved. Both shop and field cutting was required of the

fabricated caisson. Prior to dewatering, the void between the skin-plate walls was filled with tremie concrete and allowed to gain strength. A seal was formed at the downstream perimeter of the caisson to allow it to fit tightly against the existing nose of the pier. The remaining portion of the caisson was then dewatered and twenty-eight 25 mm diameter horizontal post-tension anchors and 6 vertical post tensioned anchors were installed. These were installed to secure the new pier extension to the existing pier.

The anchor system was designed to take full hydrostatic loading when an adjacent bay was dewatered. Anchors were stressed in two stages, and testing was in accordance with PTI standards. To verify that the design of the caisson would perform as intended, a finite element method of analysis was performed using ANSYS software. Design stresses of the double-walled, tremie-filled caisson were kept low to allow the contractor quicker access to the interior of the caisson and to provide a degree of conservatism. **To the authors' knowledge, this is the first time that a caisson-type system has been used to extend a pier on an intake structure of this size.** Figure 1 shows the caisson and anchor system.

The remaining piers and wing wall extensions were also designed as caissons and will be post-tensioned to the existing structure this year.

Intake and Access Service Bridges - To gain access to the new intake structure, an approach ramp will be constructed. A new 4.9 m wide by 27.4 m long double bulb-tee access bridge spans to the intake structure. Three m wide pre-tensioned, shop-fabricated concrete beams will span between piers and form the operating deck of the new intake. The bridge slabs will be welded together to form a continuous deck between pier supports. Blockouts will be formed in the precast deck beams to accept rails for a new trashrake.

POWERHOUSE BRIDGE CRANES

As part of upgrading the powerhouse, the existing 22.7 metric ton bridge cranes for the access floor and generator floor were examined to determine if they could be economically upgraded to accommodate the new turbine/ generator units. The new generator stator is the heaviest piece of equipment (25,700 kg), so with the estimated weight of the lifting beam and associated rigging, a 31.7 metric ton Class crane was selected.

The cost of upgrading the old cranes was compared to the costs associated with purchasing new cranes, as well as operation, maintenance and reliablility of new versus old. Based on this assessment, the decision was made to purchase two new 31.7 metric ton bridge cranes.

To accommodate the new cranes, a careful review and analysis of the existing crane support structures was performed. The results of the survey and analysis indicated that an extensive amount of upgrading and strengthening of the the crane rail support structures would be required.

To strengthen the concrete corbel in the generator bay, a series of post-tensioned anchors was installed at support points. In order to provide discrete support points, the rail was raised using a standard tube section. In the access bay, support points located every 0.7 m along the upstream concrete ledge were also strengthened with post-tensioned anchors. To strengthen the existing steel runway girders in the access bay, stiffener plates and "C" sections were welded to the existing girders. In the generator bay, a "W" section was welded to the top of the rail support girders to raise the rail. The existing riveted structural columns were found to be adequate. Modifications were required at the runway-to-girder column connections for strength.

TURBINE/GENERATOR

Structural Modifications - To allow for the installation of the new hydrogenerating units, structural modifications are required in the turbine wet pits and the generator floor. To allow the new steel draft tube liners to be inserted, concrete removal around the existing draft holes is required, and the annulus between the liner and existing concrete filled with non-shrink concrete.

At each of the new generators, concrete removal is required to provide proper clearance for the generators and new exciters. Since minimum reinforcing steel is present near the floor surface, no construction problems associated with concrete removal are anticipated. Unreinforced concrete thickness in the area of the generators is about 2.4 m and spans 4.6 m to the buttresses. Current support to the generators is through arch action where vertical loads are transmitted to unreinforced buttresses, 3.6 m thick. Because the new generators weigh about 40 percent more than the old ones, two parallel slots in the existing floor slab beneath the generator support assemblies will be excavated between the buttresses. W section beams will be installed that will directly transfer generator loads to the buttresses. To ensure that loads would not be transferred to the arch/slab, and only to the buttresses, a Styrofoam pad will be placed beneath the beam between the bearing assemblies.

New Hydrogenerating Equipment - Turbine / generator units 3 and 4 will be replaced entirely, including controls, governors, exciters and voltage regulators. The new capacity for each unit is approximately 10 MW. Voltage will change from 2.3 kV to 13.8 kV. New switchgear will also be installed inside the powerhouse. As part of the company's transmission upgrade the existing 60 kV substation will be replaced with a new 115 kV substation. All components will be located just west of the powerhouse. This will necessitate removing a water tank and three of the existing operators cottages.

SPILLGATE (PENDING POOL RAISE DECISION)

A rubber fabric spillgate will replace the existing 3 m high flashboard system. This will be the second gate installed in the U.S.A. that is 4.6 m high and 68.6 m long. The anchor system will consist of embedded beams that are post-tensioned to the underlying concrete. The hydraulic discharge capacity of the spillway will not be significantly affected. The controls and blower system, used to inflate the rubber fabric dam, will be housed in the powerhouse in the switchgear bay.

CONSTRUCTION MANAGEMENT

PROJECT SCHEDULE

A detailed project schedule was prepared which contains over 300 design, environmental, permitting, and construction activities. It is updated quarterly, or more often when major changes occur. The schedule separates out activities associated with each of the construction and procurement contracts for ease of use. The project schedule logic was formulated to keep the existing units generating as much as possible to minimize down time, and considers projected available river flow to better schedule construction activities. It is believed that by preparing a detailed project schedule, a better/earlier definition of critical items will save construction dollars, conflicts, and unrealistic estimates. The schedule has already allowed WWP to better control and manage the project and to direct, early-on, changes in the progression of the work. The final projected completion date for the rehabilitation project is March 1995. If WWP proceeds with the pool raise, this portion of the project would be completed by January 1996.

CONSTRUCTION CONTRACTS

WWP has elected to provide overall project and construction management itself. The contracts are a mix of both procurement and construction type contracts. This will entail overall management and coordination of 14 separate contracts. WWP has adopted this strategy because:

1) It allows economical procurement of new equipment with long lead acquisition time.
2) Some specialized equipment can be installed by WWP crews.
3) A larger cross-section of contractors will be able to participate in the project.
4) It will allow the WWP management team more flexibility in managing the project.
5) Value engineering techniques can be utilized at different phases of the project to assure the most efficient use of capital.

Procurement contracts:

Contract No. 1: Procurement of turbine/ generator units 3 and 4, including governors, exciters and voltage regulators.
Contract No. 2: Procurement of generators for units 1 and 2 (pending pool raise).
Contract No. 3: Procurement of rubber spillgate (pending pool raise).
Contract No. 4: Procurement of two 35 ton bridge cranes. WWP crews will install.
Contract No. 5: Procurement of switchgear.
Contract No. 7: Procurement of intake gates and hoists and bulkheads.
Contract No. 8: Procurement of the intake caisson.
Contract No. 11: Procurement of trash rake.

General Construction Contracts:

Contract No. 6A: Installation of the intake gates, hoists and structure including access bridge.
Contract No. 6B1: Installation of turbine/ generators and other related equipment including switchgear. WWP crews will install new controls, etc.
Contract No. 6B2: Installation of rubber spillgate (pending pool raise).
Contract No. 6C: Installation of generators for units 1 and 2 (pending pool raise).
Contract No. 9: Installation of crane rail modifications for the new bridge cranes.
Contract No. 10: Installation of the intake caisson procured under contract no. 8.

CONCLUSION

Extensive studies were conducted to determine the optimum plan for modernizing and upgrading the Nine Mile HED project. Results showed that it is economically feasible to re-power and upgrade the existing generating capacity extending the service life of the plant an additional 50 years. Based on these studies design and construction was commenced in 1992. The rehabilitation project will provide annual energy increases 23% over existing which is enough energy to serve 1,800 more homes. The pool raise portion of the project, if implemented, will raise peak capacity another 2 MW and provide annual energy increases 20% over the rehabilitation project. This will provide energy to serve 1,700 more homes than the rehabilitation project.

UNIT 1 AND 2 REHABILITATION AT CRESCENT AND VISCHER FERRY

Mohammed I. Choudhry, P.E.[1]
Michael F. Nash, P.E.[2]
William Stoiber, P.E.[3]
Andrew C. Sumner, P.E.[4]

Introduction

In 1984, the New York Power Authority was licensed to expand the generating capacity at the Crescent and Vischer Ferry Powerplants. In 1990, this expansion was completed and two new 3.1 Mw vertical Kaplan units were put in operation at each site. The Authority then rehabilitated and modernized the two original 2.8 Mw vertical Francis units at each plant. The object of the rehabilitation effort was to completely refurbish the plant mechanical and electrical equipment and to install remote operating capability. This paper discusses the rehabilitation effort.

Background

Crescent and Vischer Ferry Dams were constructed on the Mohawk River near Albany, NY, in 1912 and 1913 as part of extensive modifications to the original Erie Canal System. The two overflow spillway dams were topped with removable wooden flashboards, 12 inches at Crescent and 27

[1]Sr. Electrical Engineer, New York Power Authority, 123 Main Street, White Plains, NY 10601.

[2]Project Engineer, New York Power Authority, 123 Main Street, White Plains, NY 10601.

[3]Sr. Mechanical Engineer, New York Power Authority, 123 Main Street, White Plains, NY 10601.

[4]Construction Superintendent, New York Power Authority, Crescent Hydroelectric Project, P.O. Box 4519, Half Moon, NY 12065.

inches at Vischer Ferry. In 1925, twin 5.6 Mw hydroelectric generating stations were installed at Crescent and Vischer Ferry. Two 2.8 Mw, 2300V vertical shaft generators manufactured by the General Electric Co. were installed in each plant. Each generator was directly coupled to a 4000 hp Francis turbine manufactured by S. Morgan Smith. The turbines were designed to operate at net head of 26.5 feet and discharge 1500 cfs.

In 1983, the New York Power Authority obtained rights to the two generating stations from the New York State Department of Transportation. The Federal Energy Regulatory Commission then licensed the Authority to expand the capacity at the plants. The two original units at each site were shut down in 1988 to allow expansion construction. Expansion was completed in late 1990 and rehabilitation of the original units began in 1991. Upon completion in March, 1993, the total installed capacity at each plant was increased to 11.8 Mw.

Project Objectives

There were two primary objectives of the rehabilitation effort. Insofar as possible, the plant electrical and mechanical equipment was to be refurbished to provide 20 additional years of reliable operation. As is discussed below in detail, this entailed the complete replacement of the turbine control panel, switchgear, and step up transformer, complete stator rewind, reworking, and remachining of the various turbine mechanical components, and rebuilding of the intake gates. Specifications for these tasks were prepared by the Authority's Architect/Engineer for the project, Black & Veatch, Inc., Kansas City, Mo.

The second objective was to provide for automatic, remote operation. The plants are operated from the Authority's Blenheim-Gilboa Pumped Storage Power Project in Gilboa, NY, 40 some miles southwest of Crescent and Vischer Ferry. This was accomplished by expanding the Unit 3 and 4 remote terminal units to accommodate Units 1 and 2 and by designing the new control system and turbine control panel for automatic operation. The Unit 1 and 2 programmable logic controller was programmed to include an operating mode in which output is automatically adjusted to control headwater level. Under normal conditions the expansion units, Units 3 and 4, automatically control headwater. This mode was included in Units 1 and 2 to allow automatic river regulation when Units 3 and 4 were unavailable. Black & Veatch designed the new automatic control system for Units 1 and 2. Programming services were provided by Phoenix Controls, Seattle, WA.

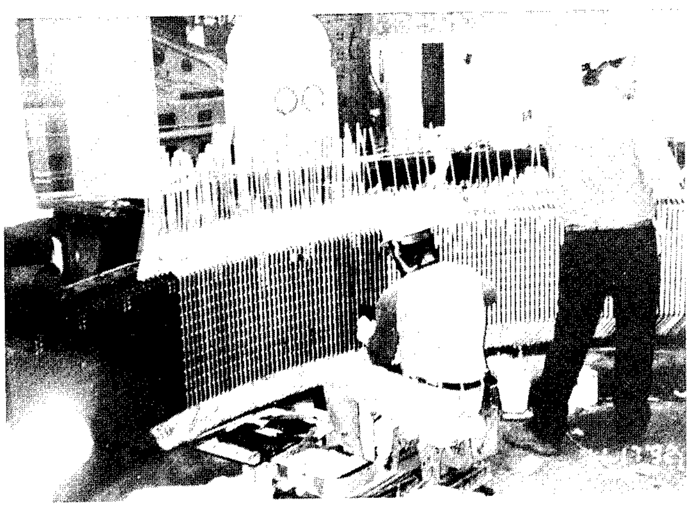
Contract Philosophy

The Authority's original approach to the rehabilitation effort was traditional, with a general contractor responsible for disassembly, testing,

mechanical shop repairs, generator rehabilitation, intake gate refurbishment, architectural modifications, and reassembly. Bids for this contract were received in November, 1990. These bids did not include stator rewind or other repairs then considered optional. The low bid greatly exceeded the budget for the work and an alternate contract approach was developed. Under this plan separate contracts were issued for intake gate and architectural modifications, generator rehabilitation, mechanical shop rehabilitation, site electrical construction and miscellaneous products and services. Authority Operation personnel disassembled and reassembled the units. Authority Project Management staff managed the overall rehabilitation effort, procured equipment and supervised construction. In all, more than 12 separate contracts were issued for the rehabilitation effort. Savings of approximately $4 million were realized under this approach.

Generator Rehabilitation

The opportunity for detailed evaluation of the Unit 1 and 2 generators was not available until Vischer Ferry Unit 2 was disassembled in 1991. The self-ventilated, air cooled generators were rated 3500 kVA, 0.8 PF, 3-phase, 60 Hz, 2300V, 90 rpm. The structural components of the stator, rotor and exciter were in good condition. However, testing showed that the electrical components needed significant overhaul. On the basis of the



Stator rewind.

Vischer Ferry Unit 2 inspection, it was decided to rewind all four generator stators. It was also decided to determine the extent of rotor and exciter repairs after disassembly and evaluation of the individual units. AC Equipment Services, now Siemens, W. Allis, WI, was selected to rehabilitate the generators.

After stripping the original coils, EL-CID tests were used to detect damaged areas of stator cores. Defects were ground out and refilled with epoxy. Great care was taken to ensure that all laminations around the repair area absorbed the epoxy and had proper coating for insulation. The effectiveness of the repair was confirmed by post-repair EL-CID testing.

Consideration was given to both hot-pressed and sealed coils for the stator rewinds. Hot pressed coils were selected for their longevity and reliability. The design employs Double-Dacron-Glass and Epoxy "B" Stage insulation. The coils were installed in the field using bottom, center, side, top fillers, ripple springs and wedges flush with the stator core. Glass mat polyester materials were used for fillers and ripple springs.

The stator support rings were reinsulated using mica tape, glass tape and epoxy resin. The coil bracing and tying was done using polyester resin and fiberglass roving. The coil connections were made using a lap brazed joint. Brazed ribbon and brazing rods were employed in incandescent-carbon brazing process. Brazed joints were insulated using mica tape, epoxy resin, and glass armor tape. A program of surge and hi-pot tests was conducted during the rewind process. Final Hi-Pot and surge tests were performed after full stator rewind.

After detailed inspection and testing, it was determined that both Crescent rotors should be rewound. Overhaul alone was sufficient for the two Vischer Ferry rotors. For rewinding, the poles were dismantled and shipped to the shop where they were rewound using the original copper. At the site, the poles were re-installed, the main leads changed, and painted in place on the rotor. The poles were mounted such that similar weight poles were placed 180° apart for balance.

The Vischer Ferry rotor poles were cleaned using environmentally acceptable solvents and cleaners. The poles were then painted and the rotor leads were changed. All pole bolts were checked and retorqued to required values.

The four exciters were shop inspected and tested to determine repair requirements. All armature windings were acceptable electrically. However, the collector rings were found rough and pitted. New rings were manufactured for three armatures. The other ring was remachined. On the field cage, four series poles were shorted, all interpoles had loose insulation

between turns, brush-rigging was dirty, rusted and had deteriorating insulation, and the field cage outer bodies had chunks of rust and old paint. All four field cages were sandblasted, and painted. All series and interpoles were rewound; and brush-rigging was partly replaced and partly reconditioned.

Mechanical Rehabilitation

Since the expansion of the two powerhouses had included the addition of stoplogs in the water passages of the older units, physical inspection of the intake gates and turbine parts was facilitated. In the absence of original drawings and calculations, physical inspection, measurement and metallurgical testing strongly influenced the rehabilitation methodology. This inspection also allowed a detailed turbine rework specification to be developed, thus encouraging a firm proposal pricing structure by numerous bidders.

Prior to any turbine-generator work, the original intake gates underwent complete shop rework. The lower half of each gate, which showed heavy rusting, was entirely replaced. The new section was bolted instead of welded to the rehabilitated section to facilitate installation and future removal for maintenance. Neoprene J-seals were added, providing a modern sealing mechanism and the screw jacks and operators were rebuilt. Since there were no provisions for removal, we are convinced that the powerhouse was built around the original gates!

The turbine parts, in particular the Francis runners which were cast of gray iron seventy years ago, were found in remarkably good condition with cavitation and structural damage minimal. The primary rework of the runners was shrinking on and pinning new stainless upper and lower wearing rings. Hymac Machine Shop, Ltd., Port Colbourne, Canada, was selected to rework the turbine mechanical parts.

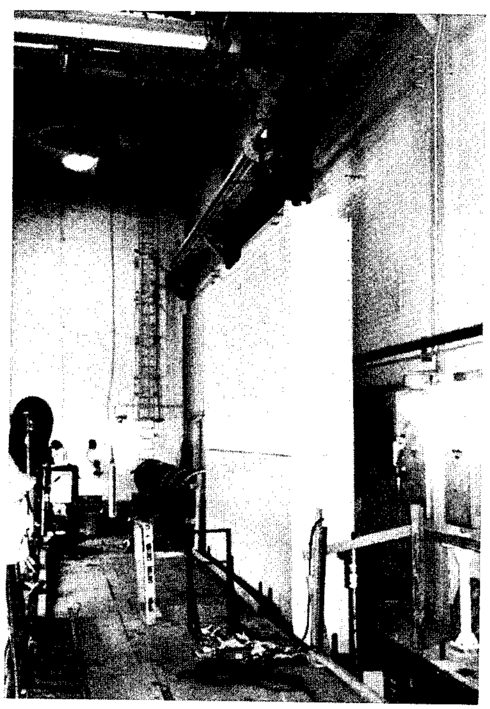
Stainless steel was employed at every bearing surface as well. New stainless sleeves were welded to the wicket gate trunnions as well as the turbine shaft at guide bearing and packing box locations; and were shrunk onto the head cover to accommodate the operating ring. New stainless panels were attached to embedded steel in the water passage for intake gate sealing and bearing surfaces.

The mechanical rebuild of the turbine, intake gates and auxiliaries was guided by environmental considerations. It was decided to remove all potential sources of waterway contamination and greased bearings in particular. Elastomeric bushings were chosen to replace the greased, bronze type at the wicket gate trunnions and all operating ring linkage points. The intake gates were refurbished as slide gates by installing new

UHMW polymer slide surfaces on the gates to both serve as guides and to overcome friction when raised or lowered.

The wicket gate trunnions were previously supported by a total of more than 16 linear inches of bronze bushing (three separate bushings) per gate. Even though this resulted in extremely low bearing pressures (less than 300 psi at maximum head), the greaseless bearing replacement segments were kept at the same lengths as the original bronze to reduce machining and fitup costs. The greaseless segments were grooved to accept O-rings, which in conjunction with an upper trunnion lip seal serve primarily to prevent silt intrusion.

Many improvements were made to components which should extend their service life. While the abundant use of stainless steel and modern bearing materials added significantly to the rehabilitation cost, these



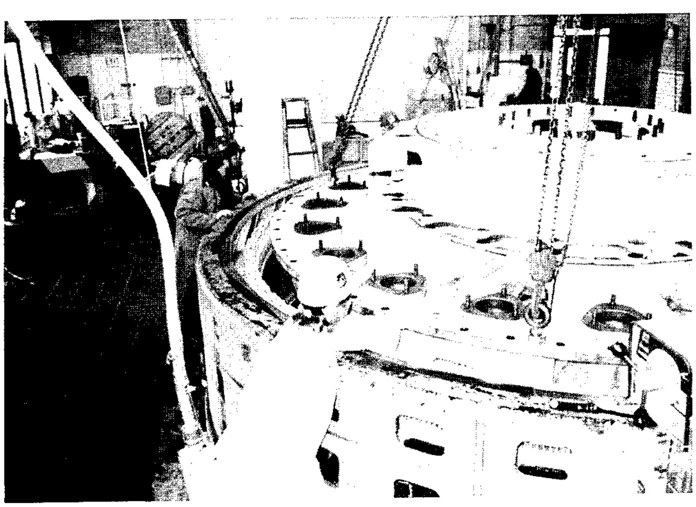
Installation of the new intake gate.

components are expected to pay for themselves in maintenance savings. Dozens of grease fittings at the old bronze bushings were eliminated, reducing maintenance requirements. Additionally, a new plant service water system will improve turbine guide bearing and packing box performance with a dependable supply of filtered cooling water.

The mechanical rehabilitation also included a complete governor and hydraulic control system overhaul. The actuator governors, which are entirely mechanical devices, the gear type oil pumps, and the permanent magnet generators were rebuilt by the Woodward Governor Co. in their shop and at the construction site. Instrumentation was added, particularly to enhance the remote operating capability of the unit, but the basic equipment design and configuration remained unchanged. Additionally, the system pressure accumulator tanks were rehabilitated and all piping was thoroughly flushed to ensure a high level of oil cleanliness.

Equipment

The existing switchgear, control panels, excitation and voltage regulation equipment were very old. To meet our objectives to provide automatic local and remote control of machines, refurbishment and retrofit of this old equipment was deemed to be unfeasible and replacement equipment was purchased.



Headcover installation.

Each plant was provided with a new 5-section, continuous walk-in panel Turbine Generator Control Cubicle to carry the control equipment for both units. The controls are processed through a G.E. Series Six-Plus Programmable Controller. As a special feature, operator interface terminals (OIT's) were also provided to display plant alarm and equipment status locally.

The new switchgear lineups selected were indoor metal-clad type for use in a 2400V, 3-phase, 3-wire grounded system, each using three G.E. 250 MVA Power/VAC vacuum circuit breakers.

The oil in existing 2.5 MVA, 3 single phase, 34.5 KV-2.4 KV, Step-Up Transformers in the switchyards were found to contain PCB. Therefore, new three phase 6.0/7.5 MVA, OA/FA non-PCB oil filled transformers have been purchased for installation at each site during 1993.

The existing battery units had manual start capability only and used the rotary exciter as self excitation, with a motorized rheostat to accomplish voltage controls for start and operation modes. In order to match new control philosophy of automatic unit start and operation, a new voltage regulation equipment cabinet was designed to interface with Programmable Controller, synchronization and control equipment. It provides for selectable VAR and power factor controls.

The existing battery chargers could not support the increased DC loads due to new Programmable Controllers and inverters. The battery chargers were replaced with larger output chargers.

The expansion unit RTU was designed with extra capacity to allow Units 1 and 2 to be operated remotely. The RTUs provide for control and supervision of these rehabilitated units from remote master station over the Authority's SCADA system.

Mechanical Disassembly & Reassembly

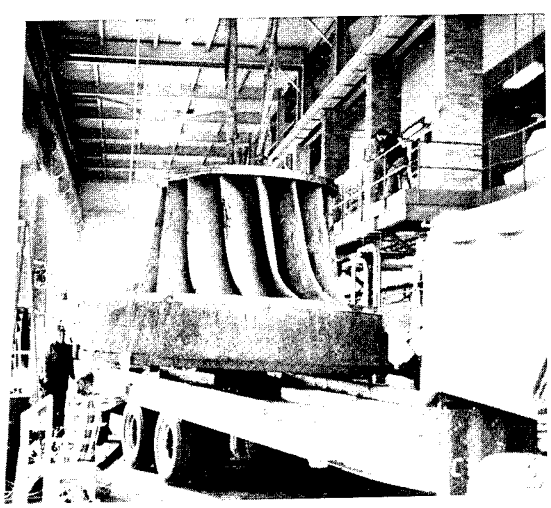
Early in the project, a management decision was made to use mechanical maintenance personnel from the Authority's Blenheim-Gilboa Project to disassemble and reassemble the turbine generators. This decision was made based on the learning benefit the mechanical crew would obtain in addition to the obvious economic advantages. Due to the extent of the work, Authority crews were supplemented by contract personnel to free Authority staff for their normal maintenance responsibilities. At times, up to six additional millwrights were added to the Authority crew. Day to day management of the work still remained under direct Authority supervision. The first turbine-generator, Vischer Ferry Unit 2, was disassembled in April, 1991. This enabled Authority engineers to determine the condition of the

major component parts of a typical unit and helped to establish the bid scope.

Reassembly of each unit had to be coordinated with other rehabilitation activities, such as generator rewinding, electrical construction, governor overhaul, piping installation and with work planned on each of the other three turbine-generators in various stages of rehabilitation. Therefore, scheduling each of the required tasks was critical to assuring efficient use of time and manpower. Time requirements to disassemble and reassemble the complete turbine-generator units varied, depending on the size of the crews and other commitments. In general, however, five man crew could disassemble a unit in about four weeks and reassemble it in eight to ten weeks.

Electrical Construction and Checkout

A separate electrical construction contract was issued for the installation and wiring of the new TGCC's, switchgear and appurtenant electrical equipment. In most respects the work was typical of utility electrical construction but the close coordination of the electrical work with



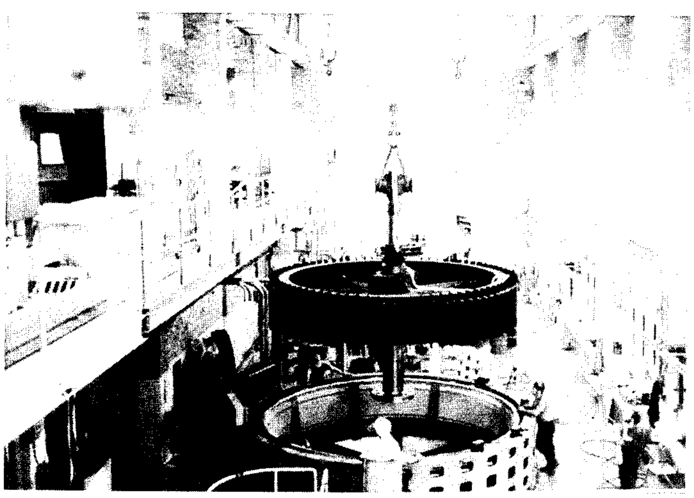
Loading runner for shipment to the mechanical repair shop.

other rehabilitation activities deserves particular mention. The unit startup schedule could be expedited only by beginning generator control wiring prior to complete reassembly. As soon as the stator was set but prior to installation of the rotor, electrical conduit installation around the stator began. This close coordination was possible only through active cooperation at the site.

Authority technicians perform a 100% wire check and selected functional tests prior to operating any new equipment. To improve the start up schedule, the Authority's checkout activities were closely coordinated with electrical construction. We also believe that checkout was expedited by having one organization's design group develop schematics for the units, turbine control system, switchgear, and appurtenant electrical system. By eliminating design interfaces the potential for design corrections and consequent delays at start up was reduced.

Control Program and Unit Start-Up

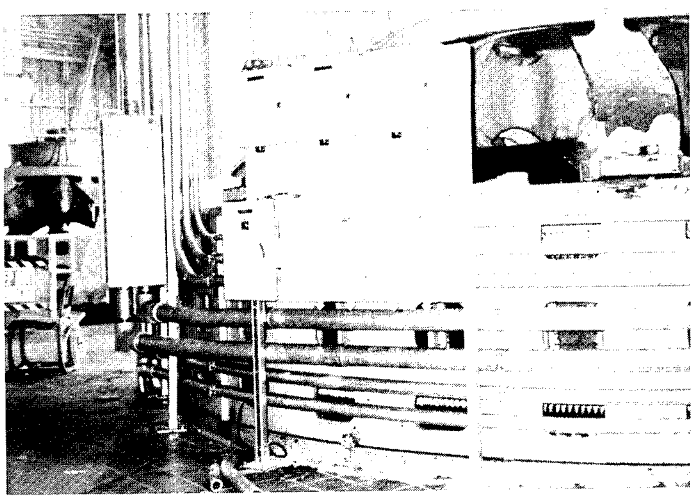
The unit control philosophy was developed with extensive Authority input. Most control functions were accomplished with the use of programmable logic controllers (PLCs) in conjunction with some hardwiring. Hardwiring was used where greater system reliability and security was



Rotor installation.

desirable. The control program was designed to accomplish three mode of operation namely "Local Manual", Local Automatic" and "Remote Control".

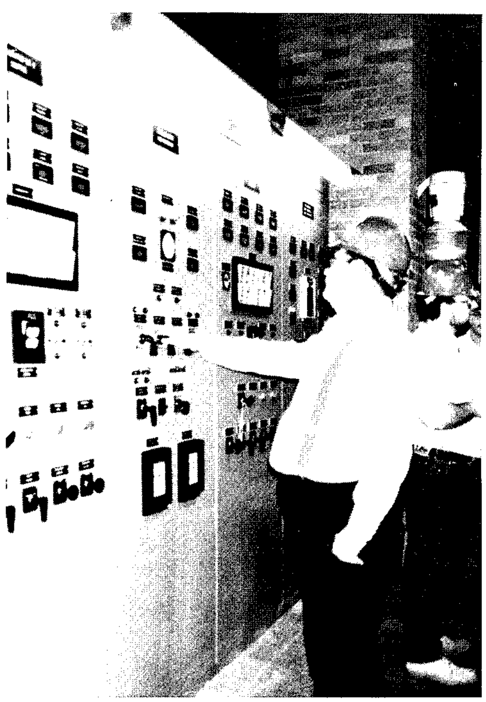
Under "Local Manual" mode, "Normal" and "Blackstart" operations are allowed. The "Normal" mode provides the units with capability of operation keeping most equipment in manual control. The unit synchronization is achieved by the use of manual voltage regulator and wicket gate control switches after the unit has reached synchronization speed and excitation contactor has closed. The unit breaker is also closed manually using the synchroscope. The "Blackstart" operation is only provided to meet Barge Canal power needs when the system grid is shutdown. In this mode, all line breakers are tripped to isolate the switchyard from the grid system. Only one unit is required to supply this radial load. The selected unit can be started with station battery only. After start command, the PLC will bring the unit up to synchronous speed; the excitation breaker will close to allow manual closing of generator breaker. At this point, the auxiliaries will be energized and normal radial operation will resume. Since the unit governors are speed control type, the synchronous frequency and load stability is maintained.



New conduit on generator stators for control cabling.

In "Local Automatic" Operation, the units are designed to maintain the preselected headwater level (HWL) while the unit loads are adjusted automatically by the PLC. The lead and lag units are pre-selected by operator. The HWL operation is only required when expansion units are either unable to run on HWL or are shutdown for repair and maintenance. In this mode, after start commend, the units go through complete start sequences up to breaker closing automatically. The units are then loaded and controlled through the automatic HWL control.

In Remote Mode" the unit is controlled automatically from a remote station over SCADA system using phonelines. Both "Remote Manual" and "Remote Automatic" mode operations can be accomplished from master station. In both modes, a start command will start the unit automatically, up to synchronization and generator breaker closure. In "Remote Manual"



Restart of Vischer Ferry Unit 2.

mode, the unit loading and VAR/power factor can be adjusted by pulsing the wicket gates and the voltage regulator system. In "Remote Automatic" mode the unit is placed under the automatic HWL control after breaker closure, and loads are adjusted automatically. The SCADA system provides the capability of adjusting the HWL setpoint from remote station.

Conclusion

As of the date this paper was authored, February, 1993, one of the four units has been restarted. Vischer Ferry Unit 2 has been operating since October 23, 1992. The unit has been controlled remotely since November 16, 1992. The restart of the remaining 3 units is planned for March, 1993.

REHABILITATION OF
UNIT #2 AT
BENNETTS BRIDGE HYDRO

Paul A. Bernhardt[1]

ABSTRACT

Niagara Mohawk Power Corporation undertook the rehabilitation of Unit 2, a 7.2 MW hydraulic turbine generator, at Bennetts Bridge in 1990. This unit was originally installed in 1914; its runner, wearing rings, and wicket gates were in very poor condition and in need of repair or replacement. In addition, testing of the generator stator and rotor indicated questionable insulation integrity, leading Niagara Mohawk to schedule rotor reinsulation and stator rewind during the turbine rehabilitation outage.

The purpose of this paper is to outline the rehabilitation project including background information, reasons for performing the project, economic analysis, history through the course of the rehabilitation, index testing method, and index test results.

NOMENCLATURE

Q	Flow rate, M^3/sec. (ft.^{3}/sec.)
K	Relative flow coefficient, dimensionless
D	Pressure differential, cm (inches)
KW	Power output of generator, kilowatts
H_G	Gross head, M. (ft.)
HWL	Headwater elevation, M. (ft.)
TWL	Tailwater elevation, M. (ft.)
PT	Potential transformer ratio
CT	Current transformer ratio
MW	Power output of generator, megawatts
H	Head, M. (ft.)
P	Power output of generator, kilowatts

[1] Mechanical Engineer, Niagara Mohawk Power Corporation, Hydro Design D-1, 300 Erie Blvd. West, Syracuse New York 13202

INTRODUCTION

The Bennetts Bridge Hydro Development was constructed in the year 1914 originally for 25 cycle generation. The four turbine-generator units were converted to 60 cycle generation between 1932 to 1966. The plant consists of a 74.4 meter (244') long concrete dam, intake tunnel, 2,384 meter (7,820') long fiberglass pipe (replaced original woodstave pipe), 364 meter (1,194') long steel pipe, and a 56.4 meter (185') tall surge tank, supplying a manifold that distributes water through four butterfly valves to four penstocks and four turbine generator units.

Unit 2 at Bennetts Bridge is a horizontal Francis (double discharge) turbine with a direct-coupled synchronous generator and shaft exciter. The configuration of the unit prior to the rehabilitation is given below.

GENERATOR:

Manufacturer:	Westinghouse, Serial #8088723
Rating:	7,500 kVA, 0.85PF, 6,375kW, 360 rpm, 12,000 volt, 3 ph, 60Hz.

TURBINE:

Manufacturer:	Wellman-Seaver-Morgan
Runner:	Double Francis, 104.8 cm (41.25") throat dia., 7,159 KW, 74.7 m. (245') net head. Horizontal 360 RPM, height of distributor 33.66 cm. (13.25")
Governor:	Woodward HR, 4149 kg.-lbs. (30,000 ft-lb), 1,378 MPa (200 psi)
Gates:	16 wicket gates.
Draft Tube:	Elbow, double discharge
Vacuum Breaker:	None
Relief Valve:	61 cm. (24") synchronous bypass, governor operated
Flow:	Best Gate (Old): 11.23 M^3/sec. (396.5 cfs) (New): 11.95 M^3/sec. (422 cfs)

EXCITER:

Manufacturer:	Westinghouse, Serial #8088822
Rating:	250 volts, 232 amps, 58kW direct connected

HEADWATER LEVELS:

Maximum - 285.6 m. (937.0 Ft.)
Minimum - 280.4 m. (920.0 Ft.)
Normal - 283.5 m. (930.0 Ft.)

INTRODUCTION (Continued)

TAILWATER LEVELS:

Maximum - 199.3 m. (654.0 Ft.)
Minimum - 198.1 m. (650.0 Ft.)
Normal - 198.7 m. (652.0 Ft.)

EXISTING CONDITIONS

The original decision to rehabilitate Unit No. 2 was based on the condition of the existing turbine runner. The runner, of cast bronze construction, was severely pitted due to cavitation. In addition, fatigue cracks were abundant as well as missing sections of buckets (blades), primarily in the inlet section. The runner seal clearances were quite large, in the neighborhood of 0.25 cm. (0.10"). It was clear that this original runner's life expectancy was exceeded.

The riveted steel spiral case was in very good condition. The seal rings had several holes (due to cavitation) and were well worn.

The wicket gate seal surfaces (top and bottom) and stems were quite worn. The stems had been spray metallized under a previous repair; the metallizing was flaking off. The wicket gate linkage had considerable lost motion, indicating the need for refurbishment.

The generator stator, rotor and exciter had been visually inspected by a consultant in 1988. Its rotor pole piece insulation was dry and brittle and some degradation was determined during recent tests. The stator, rewound in 1970, was 20 years old. The shaft exciter was in working condition, but the insulation was reported by the consultant to be mummified. There was brittle insulation coming off on three (3) stator coils. The generator was in need of complete refurbishment.

The governor was in good condition. There was some leakage through the spool valve at no load. As determined in a 1914 efficiency test, unit efficiency dropped above 80 percent gate and, above 90 percent, the output power dropped off dramatically. It was recommended to block the gates physically in the servomotor at 80 percent based on the tests.

The maximum wicket gate torque at full opening for an increased capacity unit was calculated and found that it should not increase more than 10% over the existing torque. Earlier calculation indicated that each gate requires 230.5 kg. - meters (20,000 in-lb) of torque at the stem to keep the gate closed at the current horsepower level. According to available records, the governor capacity is 133.4 kN (30,000 lb) thrust which provides 585.5 kg. - meters (50,800 in-lb) of torque at each stem. Therefore, it was possible to achieve an increase in horsepower without replacing the governor servomotor.

EXISTING CONDITIONS (Continued)

A September, 1946 letter in the correspondence files indicated that the wicket gate stems were overstressed. It was decided at that time not to replace the gates since installation of new breaking links would prevent full governor pressure being exerted on one gate stem. Calculations indicated a gate stem safety factor of 1.17 in fatigue and 3.43 for static loading based on the load capacity of the breaking links.

Leakage through the wicket gates totaled approximately 1.4 M^3/sec. (50 cfs) for the entire station. Unit 2 leakage was approximately 0.48 M^3 sec. (17 cfs). There was no operating problem with this leakage since unit overspeed upon shutdown could be prevented by the governor. Lost generation was minimal since the downstream fishery requires 0.57 M^3/sec. (20 cfs) of flow, with possible future requirements for 1.4 M^3/sec. (50 cfs) (that is, without the leakage, water would have to spilled to meet the flow requirement). Leakage could have been decreased by replacing the wicket gates with gates with larger stems (to decrease angular deflection and gap gate to gate). It was not considered cost beneficial to purchase new gates and modify the scroll case accordingly.

The spiral case is stiffened by stay rods, where use of vanes is the normal practice. The rods are unusually poor hydraulically, causing an approximate headloss of 1.22 meters (4'). These rods could have been retrofitted with airfoil shaped cowlings which would greatly reduce drag and increase unit efficiency by about 1 percent. It was determined that this modification was not practical since a large amount of field labor would be required.

REHABILITATION WORKSCOPE

A consultant was retained to determine the best approach for rehabilitation of this turbine/generator. The rehabilitation of Unit 2 was intended to extend the life of the total machine (turbine, generator, exciter). In addition, Niagara Mohawk hoped to increase the efficiency and capacity of the unit.

The focus of the consultant's study was how much additional water flow could be passed through the turbine, increasing its capacity. The consultant found that the discharge area of the runner could be increased significantly by modifying the head cover and bottom ring, thereby increasing flow capacity (see Figure No. 1). Note that this runner has 2 discharge sections, discharging to two draft tube elbows. The replacement runner would have longer bands and buckets to reduce cavitation.

The turbine shaft, generator shaft, and all support bearings were analyzed utilizing finite element analysis. The stresses in the turbine shaft were found to be unacceptable at the maximum horsepower of the new runner. The turbine shaft was, therefore, scheduled to be replaced with the runner. All pertinent critical speeds of the shafting system were calculated and found to be more than 25 percent above the maximum runaway speed, indicating that resonant vibrations would not occur.

REHABILITATION WORKSCOPE (Continued)

It was determined that the increased output of the turbine could be absorbed by the generator if the stator was rewound, allowing more copper area in each slot and a better grade of insulation. In addition, the generator rotor and the excitor would need to be reinsulated to the same insulation class as the stator to match the rewound generator's capacity.

Niagara Mohawk decided to proceed with the upgrade as specified by the consultant, who projected a 30% capacity increase and an 11 to 16% efficiency improvement.

In addition to replacing the runner and modifying the head cover and bottom ring, the following major turbine components were replaced: turbine shaft, wicket gate bushings, wicket gate linkage and pins, and runner seal plates. The following components were refurbished: wicket gates, and operating ring; governor regulating shafts, packing collars, connecting rod pins; and turbine bearings.

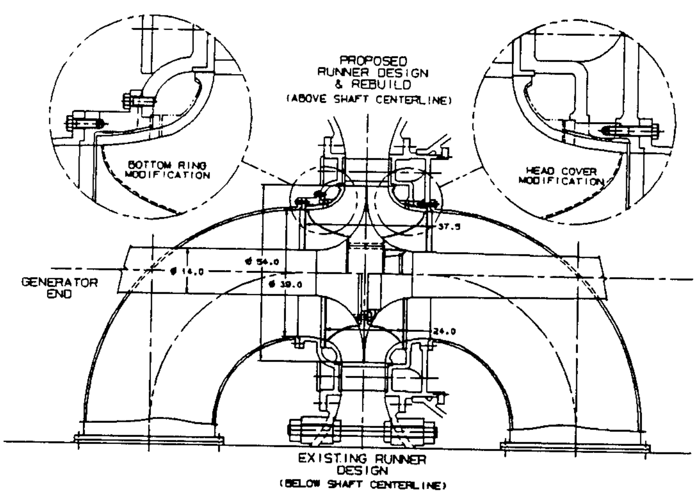


Figure 1

PROJECT HISTORY

Niagara Mohawk specified and managed the upgrade project, letting separate contracts for the turbine and generator work. The unit was disassembled during a low water period, allowing turbine parts to be send out to the turbine vendor for rehabilitation/replacement, and giving access to the generator for the rewind contractor to begin his work.

PROJECT HISTORY (Continued)

The new runner and its seal rings are stainless steel for cavitation resistance. The wicket gate seal surfaces and stems were overlayed with stainless steel weld metal to provide new, harder wear surfaces.

The generator stator was rewound to a complete Class F insulation system. RTD devices were installed for temperature monitoring of the windings. The rotor was reinsulated to a Class F system. The exciter was reinsulated and refurbished to support the generator nameplate increase.

One of the difficulties envisioned during the planning process was dealing with the 32-ton flywheel coupled between the turbine and generator shafts. The flywheel was blocked in place and fortunately presented no major problems during reassembly.

One of the cast iron draft tube elbows was damaged during initial disassembly of the turbine. A piece of the elbow was broken out. It was determined that it was more economical to repair than to replace the elbow. A proprietary process called "Metalstitching" was used to effectively join the broken piece to the body of the elbow.

Unfortunately, the turbine vendor was late due to the large amount of welding and finish grinding required on the new runner. This delay in schedule allowed the headpond to fill to the point where spilling occurred for several days before the unit could be put back on line.

ECONOMIC ANALYSIS

The following is a compilation of the costs associated with the rehabilitation of Unit 2.

	(In $)
Turbine Contractor	518,000
Generator/Exciter Contractor	273,000
Misc. Material	53,000
Misc. Expenses	8,000
Field Labor	115,000
Engineering & Administration	46,000
TOTAL DIRECT COST	**$ 1,013,000**

The index test, as reported subsequently, indicated a 32 percent increase in capacity after the rehabilitation was complete. The benefit/cost ratio for this project, levelizing this increased capacity benefit over 25 years, is 1.84. (This verifies the original benefit/cost calculation used to justify the rehabilitation).

INDEX TESTING METHOD

An index test was performed in accordance with IEC (International Electric Committee) 41 with the exception of power and flow measurements. An air over water manometer was constructed of readily available (hardware store) materials for the differential pressure (relative flow) measurement. The power was measured using a calibrated watthour meter and a stopwatch to time a certain number of dial revolutions.

Index tests involve the measurement of a pressure differential (D) in the water passages of the turbine which is proportional to the square of the turbine flow. The square root of this differential pressure represents relative flow, and the power output divided by the square root of D represents relative efficiency.

$$\text{Relative Flow} = Q = K\,D^{1/2}$$

$$\text{Relative Efficiency} = \frac{P\ [kw]}{D^{1/2}}$$

Index testing verifies the improvements in capacity and efficiency, as opposed to the actual unit efficiency. This testing was done at a low cost to demonstrate that an elaborate, costly testing program to measure absolute efficiency is not always necessary.

TEST MEASUREMENTS

During the tests the following measurements were made.

a. Wicket Gate Servomotor Stroke

Wicket gate servomotor stroke was measured with a machinist's scale attached to the servomotor.

b. Headwater Level

Headwater level was measured with the existing rotating tape which is downstream of the trashracks in the inlet structure to the pipeline. The trashracks were clean and there was no trash in the headwater. Elevations were based on a 1980 survey (that is, a decision was made to not re-survey the benchmark).

c. Tailwater Level

Tailwater level was measured using a scale and a float. The scale was attached 0.61 meter (2') above the float's waterline. As such, 0.61 meter (2') was added to each scale reading (see Figure #2). Elevations were again based on the 1980 survey.

TEST MEASUREMENTS (Continued)

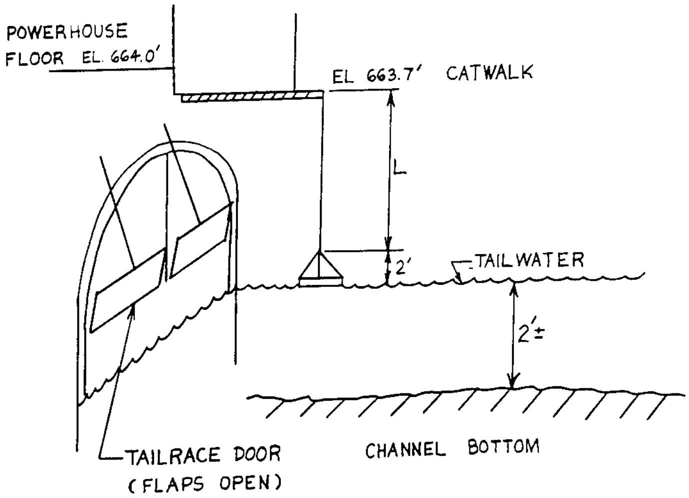


Figure 2

d. Gross Head

This is defined as the headwater level minus the tailwater level.

e. Differential Pressure D

Differential pressure between the outer and inner surfaces of the spiral case was measured with piezometers per the Winter-Kennedy method for determining relative discharge. Because piezometer taps per Figure 46 of I.E.C. 41 were unavailable, a tap was used which included copper tubing and Devcon compound to obtain a flush inside water passage. The differential pressure from these relative flow taps was measured with a differential compressed air over water manometer per Figure 53 of I.E.C. 41. This manometer was built with hardware-store available materials mounted on a plywood-backed stand (i.e.: clear flexible hose, globe valves, hose clamps, etc.). The water level in each leg of the manometer was read simultaneously on steel tapes graduated in 3 mm increments (1/100 of a foot). {See Figure #3}

f. Servomotor Stroke Versus Gate Opening

The servomotor stroke was measured and simultaneously the gate opening was measured between two gates.

TEST MEASUREMENTS (Continued)

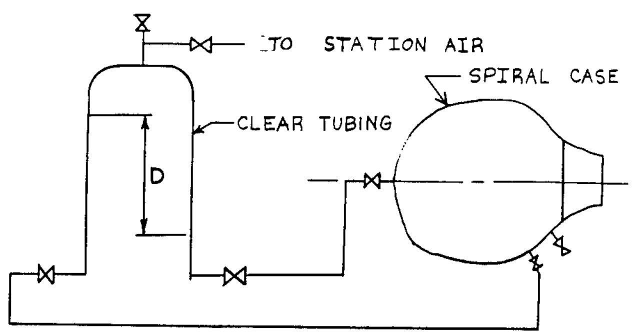


Figure 3

g. Power Output

The unit has a rotating watt-hour meter which was recently calibrated. The disk rotation was timed with a stop watch. The average KW of the unit was calculated by the following equation.

$$\text{Avg KW} = \frac{(\text{Watt Meter Disk Revs})(\text{PT})(\text{CT})(\text{KWH Meter Const.})(3600\ \text{Sec/Hr.})}{(\text{Elapsed Time in Seconds})(1000\ \text{Watts/KW})}$$

where PT was equal to 100, CT was equal to 120,
KWH Meter Constant was 1.2

$$\text{Avg. KW} = \frac{(\#\ \text{Revs})\ (14{,}400)\ 3.600}{\text{Time in Seconds}}$$

TEST PROCEDURE

The turbine was dewatered and the scroll case pressure taps were inspected on the day before the test to ensure they were clear. The intake trashracks were verified to be clean on the day of testing.

The index test was conducted at different wicket gate positions. For each test point, multiple readings were taken of various instruments and water elevations and the average value used. For each test point, the wicket gates were held in a fixed position by the governor gate limit device. During the duration of the tests, the unit was set at 1.0 power factor.

TEST PROCEDURE (Continued)

Units 1 & 3 were motoring and the wicket gates for these units were kept closed during the testing period. Unit 4 was shut down for maintenance. There was some leakage from units 1 and 3 wicket gates while motoring, but this leakage was considered insignificant.

INDEX TEST RESULTS

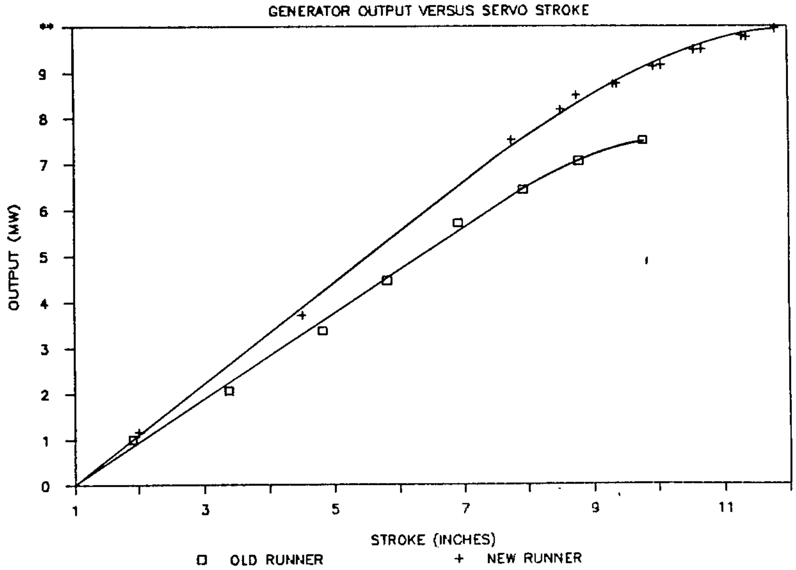
The turbine performance characteristics as determined from the index tests are presented on the following figures (English units). The characteristics have been adjusted to a reference gross head of 85.95 meters (282').

Figure 4: Generator output [kw] versus servomotor stroke

Figure 5: Relative efficiency (Generator Output in MW/ $D^{1/2}$) versus servomotor stroke.

FIGURE 4

GENERATOR OUTPUT VERSUS SERVO STROKE



EXPLANATION OF SOME COMPUTATIONS

The square root of the differential pressure is adjusted to a reference gross head of 85.95 meters (282') using the relationship:

(Square Root of D2) / (Square Root of D1) = (H2 / H1) ^ 0.5

The generator output is adjusted to a reference gross head of 85.95 meters (282 feet) using the relationship:

P2 / P1 = (H2 / H1) ^ 1.5

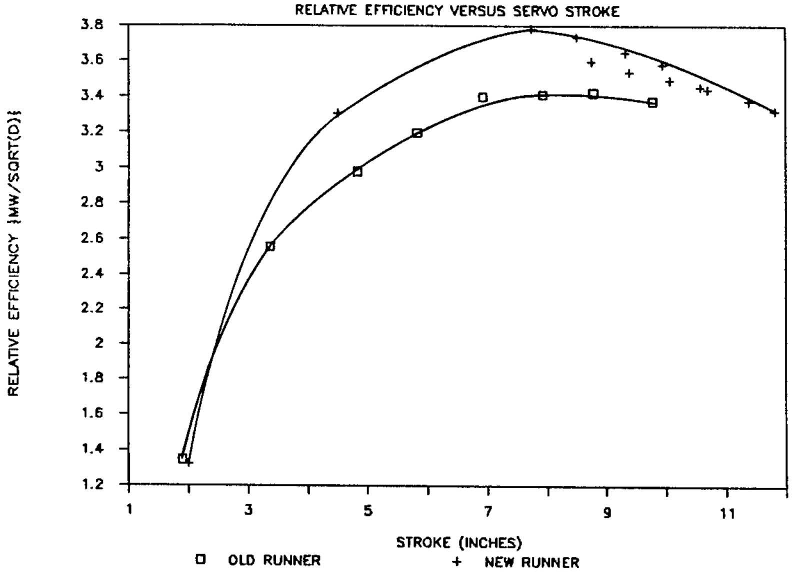
DISCUSSION OF RESULTS

The relative efficiency versus servomotor stroke at 85.95 meters (282') gross head (Figure 5) is the result of the Index Test. This figure presents the output at which peak efficiency can be expected. Further, it is also indicated that the renovated unit has 10.5 percent higher efficiency than the old unit.

$$(MW/\ D^{1/2})\ \text{renovated} / (MW/\ D^{1/2})\ \text{old} = 3.782/3.423 = 1.1049$$

FIGURE 5

RELATIVE EFFICIENCY VERSUS SERVO STROKE



DISCUSSION OF RESULTS (Continued)

In addition, the maximum capacity of the unit was increased by 32 percent.

$$\frac{9.943 \text{ MW} - 7.507 \text{ MW}}{7.507 \text{ MW}} = 0.32$$

The accuracy of the results is estimated to be within ± 5 percent.

CONCLUSION

In summary, Niagara Mohawk realized a 10.5 percent increase in unit efficiency at best gate and a 32% increase in maximum power capability as a result of the rehabilitation. A heat run test indicated the stator temperature rise at nameplate kVA is 40.3°C, well within the 60°C rise limit for Class F. It is hoped that the life of the unit has been extended by 25 or more years.

The results of this rehabilitation project encouraged Niagara Mohawk to continue with plans to rehabilitate the remaining three (3) units at Bennetts Bridge. Unit 3 and 4 rehabilitation is in progress; Unit 1 is scheduled for rehabilitation in 1995.

ACKNOWLEDGEMENTS

The turbine rehabilitation was completed by American Hydro Corporation. The generator rehabilitation was completed by General Electric. Considerable support and assistance was provided by Messrs. J. Flood, F. Howley, and M. Lachut of Niagara Mohawk.

SENECA PUMP TURBINE REHABILITATION

PENNSYLVANIA ELECTRIC COMPANY/GPU
CLEVELAND ELECTRIC ILLUMINATING COMPANY

CLAYTON C. PURDY*
CURTIS D. WATERS**

Abstract

The rehabilitation of the pump turbines at the Seneca plant was driven by the discovery of cracked wicket gates which posed a threat of a long unscheduled shutdown. Lengthy annual shutdowns were required to repair cavitation damage. Increased efficiency, decreased motor input and decreased vibrations were also factors. Field tests were conducted by the volumetric method before and after rehabilitation to determine pump, turbine and cycle efficiencies. This paper should be read in conjunction with the case study paper presented at Waterpower 91 and the field test paper being presented at Waterpower 93.

Introduction

The Seneca Pumped Storage Plant is located on the Allegheny River and Lake Kinzua near Warren in northwestern Pennsylvania. The plant is owned by the Cleveland Electric Illuminating Company and the Pennsylvania Electric Company (Penelec)/GPU. The project was designed by Harza Engineering. The plant consists of two 225 MW pump turbines and one 35 MW conventional

* Clayton C. Purdy, Consulting Engineer (retired from Ebasco) 30 Meadowbrook Road, Syosset, NY 11791

** Project Engineer, Penelec/GPU, 1001 Broad Street, Johnstown, PA 15907

turbine as well as motor generators, generator, valves and other powerhouse equipment. Newport News and Shipbuilding Company designed and supplied the pump turbines, Westinghouse the motor generators and Woodward the governors. Operation commenced about 1970. During the middle 1970's several bolts in the Unit-2 shaft coupling sheared and others loosened, allowing the runner to swing eccentrically, damaging the wearing rings and increasing their clearance. The resulting increase in thrust necessitated limiting Unit-2 to 175 MW when generating. Input to Unit-2 increased from 225 to 240 MW during low head pumping. At almost the same time the runner cone of Unit-1 became detached while in the pumping mode, damaging the blades. Repair consisted of trimming off a 6-inch width along the full length of each of the nine blades. The efficiency of both units decreased an estimated four percent in each mode. In addition there were large vibrations in Unit-2. Cavitation repair caused two to three week shutdowns for each unit annually. Prerehabilitation inspection revealed four cracks in each of the forty wicket gates where the upper stems join the blades. In this paper "runner" means the impeller-runner and "generating mode" means turbine mode.

Unusual Aspects

Unit-2 pump turbine has the capability of discharging to either Lake Kinzua or the Allegheny River which is about 100 feet below Kinzua. This is done by valving and a torus type draft tube. During the 1990 model tests Voith engineers discovered that the torus draft tube caused a swirl as the water approached the runner in the pump mode. Since the runner blades were designed for no swirl, vanes were installed in the draft tube to remove the swirl. Unit-1 always discharges to Lake Kinzua. Unit-3, the 35 MW turbine, always discharges to the Allegheny River. With this arrangement it is possible for 53 percent of the water pumped from the lake to be discharged to the river at 100 feet more head. The P/G ratio for the year can be unusually low compared to normal pumped storage plants.

Project Data

Figure 1 is a schematic of the project. Lake Kinzua, the lower reservoir, has a volume of 1,180,000 acre feet, equal to 160 days average flow of the Allegheny River or one of the pump turbines. In the pumping mode the dynamic head can vary from 660 to 820 feet. Rated input is 300,150 horsepower at 660 feet head. In the generating mode, discharging to Lake Kinzua, the maximum output is 253 MVA at a net head of 793 feet. When Unit-2 is discharging to the Allegheny River the turbine output is 392,000 hp at a net head of 860 feet.

Reasons for Rehabilitation

- Replace the cracked wicket gates.
- Reduce or eliminate annual outages for cavitation repair.
- Increase efficiencies by providing new runners and wearing rings.
- Reduce the input to the pumps.
- Reduce excessive vibrations at Unit-2.
- Replace the straight through wearing rings with labyrinth type.
- Reduce hydraulic thrust at Unit-2.
- Extend the life of the pump turbines by replacing other worn parts.
- Replace the packing type main shaft seal with a mechanical seal.
- Repair fatigue cracks in the main shaft flanges.
- Eliminate the swirl in the draft tube of Unit-2.

Specifications and Bidding Documents

After the reasons for rehabilitation were evaluated Penelec asked Ebasco to proceed with the engineering. The specifications were written in such a way as to encourage the contractor to provide the best solutions to the problems rather than to handicap him with a multitude of guarantees and penalties. The main constraints were:

- the existing water passage dimensions are not in agreement with modern pump turbine design
- the setting relative to tailwater could not be changed
- the head ranges require two runners of slightly different design
- the rpm could not be changed
- no change in motor generator ratings

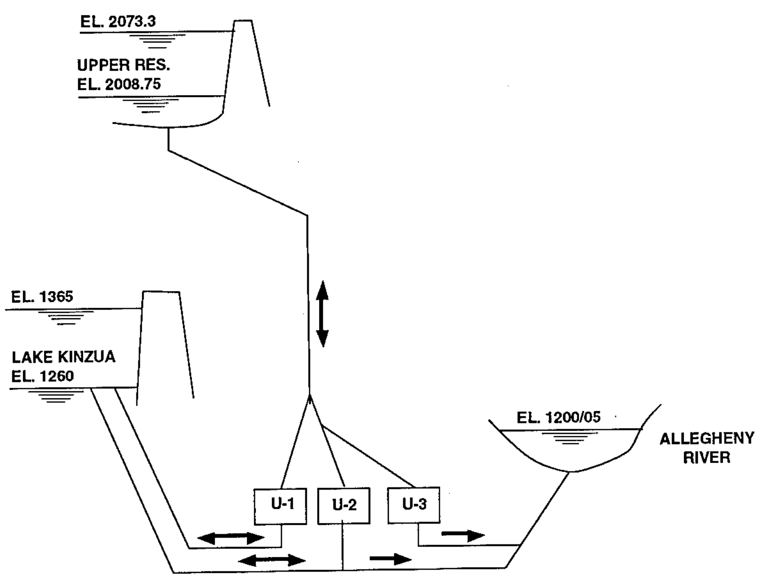
The specifications required a guarantee of pump efficiency at the lowest dynamic head but the guarantee of flow rate allowed a margin on input so future degradation would not result in overload of the motors. The authors do not agree with the triple guarantee of efficiency, flow rate and input at one point. Best results are obtained when the designer is allowed leaway and is not bound by unnecessary restrictions. Guarantees to 0.1 percent are not provable when test results are in a band of plus or minus 2 percent.

The specifications called for testing one model for each pump turbine because of the different head range and different draft tubes. Testing each prototype before and after rehabilitation was also specified. As will be explained later this prototype testing followed nearly normal operation and dispatching.

The runner material is CA6NM stainless steel. The blades, crown and band were cast separately and all water passages were machined according to an exact stepup from the model before being welded into a complete assembly. The wicket gates are A182F6NM forgings designed to prevent fatigue cracking at the upper stem to blade junction. Penelec provided their usual instructions to complete the bidding documents.

Bids were received from three sources. These bids were evaluated for technical quality and response to the specifications. Discrepancies were discussed with each bidder separately. The technical recommendation for award was based on (1) guarantees of efficiency, flow rate, input and output, (2) compliance with the specifications, (3) modern design and construction and (4) the capability of the bidder to perform all phases from design through assembly, startup and testing. Ebasco was responsible for project overview and expediting, Penelec for quality assurance.

Figure 1
Project Schematic



Expectations and Guarantees - Typical Values

	Unit-1	Unit-2
Pumping		
Expected efficiency at 660 ft TDH	91.5	91.0
Guaranteed efficiency	91.0	90.5
Model test plus 2.4*	91.8	91.1
Expected efficiency at 750 ft TDH	92.9	92.4
Guaranteed efficiency	92.4	91.9
Model test plus 2.4*	92.0	91.7
Prototype before rehab 722 ft TDH	86.6	85.0
Prototype after rehab 716 ft TDH	**	91.9
Expected efficiency at 820 ft TDH	90.8	90.3
Guaranteed efficiency	90.3	89.8
Model test plus 2.4*	90.3	90.4
Guaranteed cfs at 820 ft TDH, model test stepped up	2297	2284
Input hp not to exceed at 660 ft TDH	300,150	300,150
Model hp stepped up, not measured	300,150	301,811
Generating		
Expected efficiency at 630 ft Hn FG	87.5	87.1
Guaranteed efficiency	87.2	86.9
Model test plus 2.4*	90.7	88.1
Expected efficiency at 800/851.5 ft Hn	88.4	90.2
Guaranteed efficiency	88.1	90.0
Model test plus 2.4*	91.2	89.7
Prototype before rehab 705 ft Hn BG	86.9	82.8
Prototype before rehab 831 ft Hn BG		86.4
Prototype before rehab 700 ft Hn FG	84.1	
Prototype after rehab 705 ft Hn BG	**	88.0
Prototype after rehab 825 ft Hn BG	**	88.8
Expected efficiency, eye of chart 962' Hn	92.6	92.2
Guaranteed efficiency, eye of chart 962' Hn	92.3	92.0
Model test plus 2.4*	93.2	90.6
Guaranteed max output hp, 630 ft Hn	2,231,945	
Guaranteed hp, 851.1 ft Hn		346,155
Axial thrust before rehab, lb		1,000,800
Axial thrust after rehab, lb		226,000
Runaway before and after rehab rpm	378	378

FG = full gate BG = best gate

Unit-1 Arithmetic average of five efficiencies
Guaranteed (91.0 + 92.4 + 90.3 + 88.1 + 87.2)/5 = 89.8
*Model test increased 2.4, (2/3 of the Moody formula using 2 as exponent)
(91.8 + 92.0 + 90.3 + 90.7 + 91.2)/5 = 91.2
Expected (91.5 + 92.9 + 90.8 + 87.5 + 88.4)/5 = 90.2

**Test result not available until later

Schedule and Site Work

The specifications and bidding documents were completed by September 1989. The first unit was scheduled to be delivered in time to begin site work September 1, 1991. Startup was scheduled for December 15, 1991. The second unit was scheduled one year later. Within this framework there were various milestones. The tests of the rehabilitated units were scheduled for April 1992 and 1993 respectively. There was a delay of approximately one month in the completion of the first unit due in part to the discovery and repair of fatigue cracks in the main shaft flange. After analysis of these cracks it is estimated that the life of the shaft exceeds the probable life of the plant by several times. It is possible that these cracks were introduced during the original fabrication but were not noticed because they were so small. Similar cracks were found and repaired in the other unit.

Bolt removal from the shaft flange was unusually difficult. Some of the bolt shanks adhered to their holes almost as if they had been welded their entire length. Surfaces for the new stationary wearing rings opposite the rotating surfaces were machined in place. All wicket gate bushing holes were lined bored in sets of three by assuming that top holes were in true position. The bushings were replaced.

Performance Regime

Pumps and hydraulic turbines have been separate for many years and due to this separation their performance curves are different. Pump performance curves are based on head and input versus flow rate. Turbine performance curves are based on efficiency and flow rate versus output at each net head. Figure 2 shows a combined performance regime as used for the Seneca rehabilitation. The figure shows the controlled operation during the pump mode and the wide spectrum of operation possible during the generating mode. The figure also illustrates that the rating of 225 MW in either mode seldom occurs. The figure also illustrates that the maximum efficiency of the pump occurs every day but the maximum efficiency of the turbine usually does not occur because of load regulation.

Flow Measurement

The choice of the method of flow measurement for the tests before and after rehabilitation took into account the advantages and disadvantages of code approved methods such as current meters, salt velocity, pressure-time and ultrasonic. The current meter and salt velocity methods require apparatus to be installed inside the water passages. Shutdown and dewatering are required before and after testing. The pressure-time method is not suitable for the pump mode

because it is based on load rejection. The ultrasonic method is not suitable for the pump mode because bubbles may be formed.

The volumetric method was specified because it is ideally suited to Seneca where the upper reservoir is relatively small and its volume is known within close limits. Operation of a single unit in either the pumping or generating mode causes a rise and fall of about 25 feet in 8 hours. The tests followed nearly normal dispatching, pumping for 8 hours and generating for 8 hours at best gate, or generating for longer periods at small gate openings or for shorter periods near full gate opening.

The elevations of the upper reservoir were accurately measured as were time, input, output, gross head and conduit loss. If the reservoir volume is accurately known the individual pump and turbine efficiencies can be calculated accurately. If the reservoir volume is not accurately known the accurate measurement of the change in elevation permits accurate calculation of the cycle efficiency. Cycle efficiency is the product of pump and turbine efficiency over the same range of gross head.

Test Plan

The model tests were reported at Waterpower 91 in the paper titled "Case Study in the Upgrade and Rehabilitation Techniques of a Pumped Storage Installation" by Kepler and Franke. The prototype testing is being reported at this conference by Kepler and Dietz.

Operation and Dispatching

The relationship between head and flow during the pumping mode is controlled by wicket openings which result in the most efficient operation. The relationship between head and flow is almost unlimited during the generating mode and is controlled by the dispatcher (1) best gate, (2) maximum gate, (3) constant output, (4) system regulation or (5) combinations of these.

Input power (P) is composed of electric loss, conduit loss, pump loss and power to drive the pump. Output power (G) is composed similarly. The ratio P/G is a measure of cycle efficiency. P/G = 1.28 is theoretically possible but not usual. Some plants have P/G ratios of 1.4 or higher. At Seneca because of the extra 100 feet head when discharging to the river, P/G ratios as small as 1.16 are possible. The combined use of the three Seneca units is flexible and results in a complicated dispatch.

Figure 3 is a P/G diagram for either Unit-1 or -2 operating separately. Typical wicket openings, constant output lines and P/G ratio lines are shown. Head is the ordinate and flow the abscissa. Conduit loss is shown at the top of the figure. The lowest P/G ratio is 1.318 (all losses up to the generator terminals are

included). A low ratio is obtained only if the operator controls the wicket gates to follow the dashed line.

Conclusions

The cast steel wicket gates were replaced by forged steel wicket gates.

Inspection of Unit-2 runner after almost a year of operation shows no evidence of cavitation damage.

Efficiencies have been increased significantly.

The input to both motors is now less than rated.

Both units operate without noticeable vibration.

The straight through wearing rings were replaced by labyrinth type wearing rings.

Hydraulic thrust has been reduced to 25 percent of the former thrust.

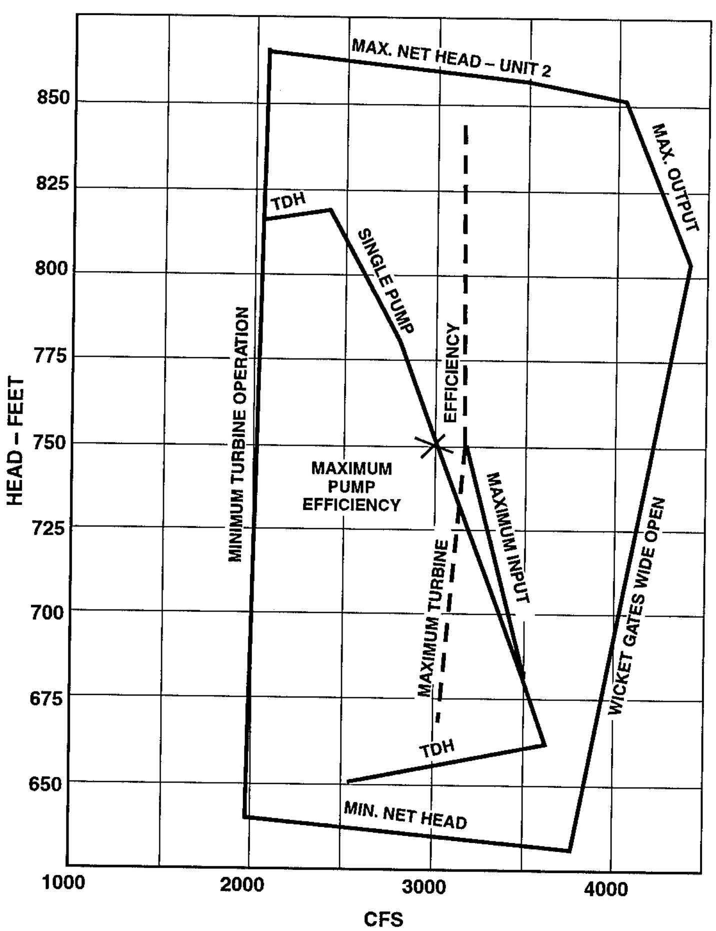
Small worn parts were replaced to extend the life of the equipment.

The packing type main shaft seal has been replaced by a mechanical seal.

Fatigue cracks in the main shaft flange were removed.

The swirl at the entrance to the Unit-2 runner during pumping was eliminated by vanes in the draft tube.

Figure 2
Performance Regime



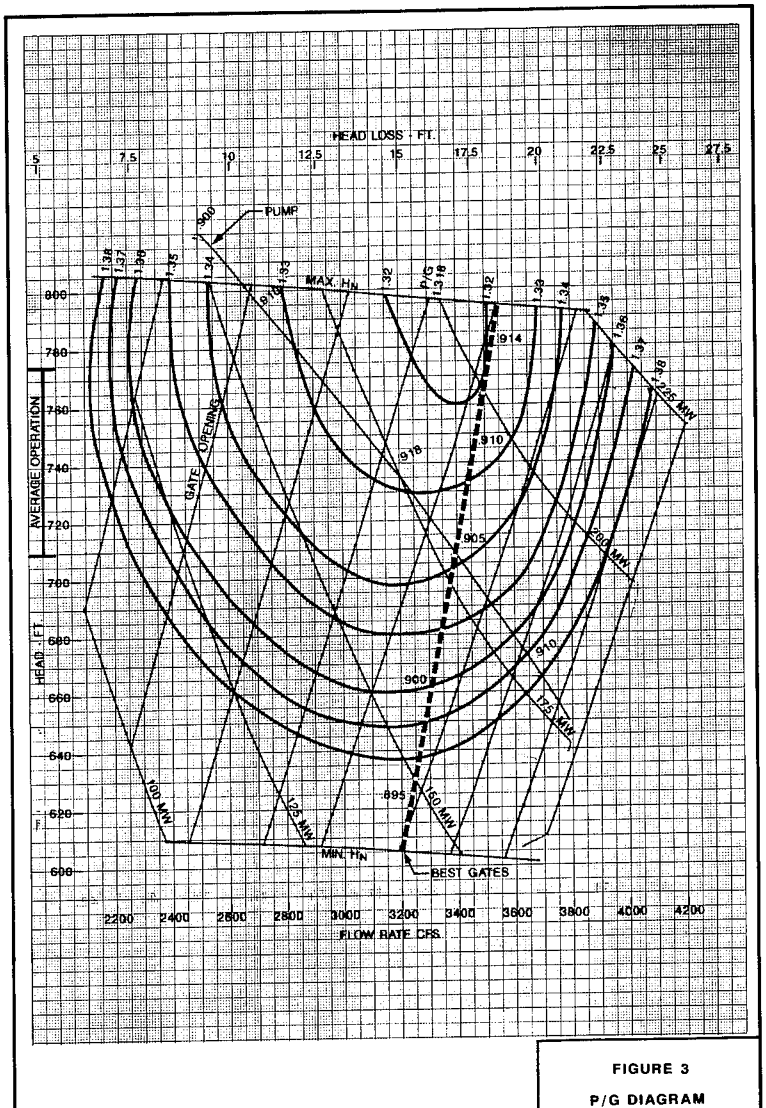


FIGURE 3

P/G DIAGRAM

SENECA PUMPED STORAGE

LUDINGTON #5
SITE OVERHAUL OF A LARGE PUMP/TURBINE

Peter E. Papaioannou *
Stephen D. Rinehart **

Abstract

Five years after the first major overhaul of a Ludington Pumped Storage Project unit, it was deemed necessary to perform a major overhaul of a second unit. The Ludington Pumped Storage units were placed in commercial operation in 1973 and have only had minor repairs every few years.

The Ludington Pumped Storage Project is located near Ludington, Michigan and is situated on the East Shore of Lake Michigan. The powerhouse contains six of the largest pump/turbines in the world and is owned and operated by Consumers Power Company.

This paper reviews the challenges unique to this particular overhaul. Since the Ludington units are the largest pump/turbines in the world, size becomes a problem due to material handling procedures and effectively using available lay-down areas. The powerhouse design presents another challenge because the generator cover is located on the roof; therefore the weather greatly contributes to the overall efficiency of the progress.

Site machining was utilized to return the unit to O.E.M. specifications. Reviewed in this paper will be the site machining processes used on Unit #5 which includes the following:

* Project Manager Hydro and Environmental Projects, Consumers Power Company, Jackson, Michigan

** Senior Project Manager, Voith Hydro Inc., P.O. Box 712, York, Pennsylvania 17405, Telephone: (717) 792-7000

1. Discharge Ring rough machining, welding and re-machining.
2. Remachining the headcover upthrust collar surfaces.
3. Line boring of the wicket gate bushing penetrations.
4. Remachining generator rotor spider "T" slots.

Shop rehabilitation of the wicket gates was deemed necessary due to the magnitude of the repair and the size of the wicket gates.

Due to the enormous scope of activities to be accomplished on an overhaul of this size, scheduling became an important tool in monitoring the outage. The schedule was duration driven and monitored the expended manhours weekly. This scheduling accuracy was accomplished by having all the scheduling hardware and software on site making it possible to update and review the schedule daily.

Introduction

This was the first major overhaul of Unit #5 since the original commissioning in 1973 and the second unit to be overhauled at the Ludington facility. During the 18 years of operation, Unit #5 has run nearly 71,000 hours and has gone through over 13,500 start and stop sequences.

The Ludington units are of the following type:

Hitachi LTD. (manufacturer of pump/turbine and generator/motor)
VTFK 3-W-RD (semi-umbrella)
325,000 KVA
112.5 RPM
64 Poles
Split, six bucket carbon steel runner, stainless steel overlay
Vertical Francis pump/turbine-generator/motor

Management Concept

The management concept utilized for this overhaul was different from the typical scenario used on other similar rehabilitations. Voith Hydro, Inc. in York, Pennsylvania, was contracted to manage the outage from the pre-outage testing up to and including the post-outage testing. Voith's management team was totally responsible for the scheduling as well as the technical direction of the crafts. Consumers Power Company and Voith Hydro, Inc. combined their resources and developed an integrated management team. The craft workers, including the first line supervision, were supplied by Consumers Power Company's Field Maintenance Services (FMS).

Scheduling

On an outage of this magnitude, scheduling becomes an important tool. Due to the amount of emergent repair identified during the disassembly phase, the schedule had to be modified to accommodate the increase in scope. The scheduling accuracy contributed to the success of the overhaul.

Because the condition of the unit was worse than anticipated many schedule revisions were required to factor in approximately 20% extra work and minimize the impact on critical path.

The site management team used this feature to quickly identify problem areas. If the manhours on a certain task were found to exceed the estimate, the attention of the team was shifted to the problem area and in most cases alleviated the impact of a serious problem.

Since the expended manhours were tracked by work task, a work breakdown structure code was assigned to each task. These codes were then tied to a numbering system used by Consumers Power Accounting System.

Pre-Outage Testing

Before the disassembly of the unit commenced, a pre-outage testing program was initiated. This program was very important to the condition assessment and provided the customer with a reference point regarding a before and after comparison.

The Pre-Outage Testing was a joint effort between Voith's on-site management and the Energy Supply Services Department of Consumers Power Company.

Disassembly of Unit

Due to the size of these units, the laydown areas to accommodate the disassembled parts had to be addressed prior to the start of the outage. The overhaul Management Team met with Ludington plant management to discuss various aspects of available laydown area. All of the laydown area is outside on top of the powerhouse. Consequently, the weather became a contributing factor to the efficiency of the cleaning and storage of components. For example, the rotor poles were taken to a nearby warehouse for rehabilitation. The floor loading was investigated and the placement of certain components was dictated according to weight along with the degree of weather protection required.

Another concern the management team had to face was the powerhouse design. The generator cover is located on the top of the powerhouse as is the overhead crane. The weather presents problems unique to this type of design. Figure 1 shows the top of the powerhouse.

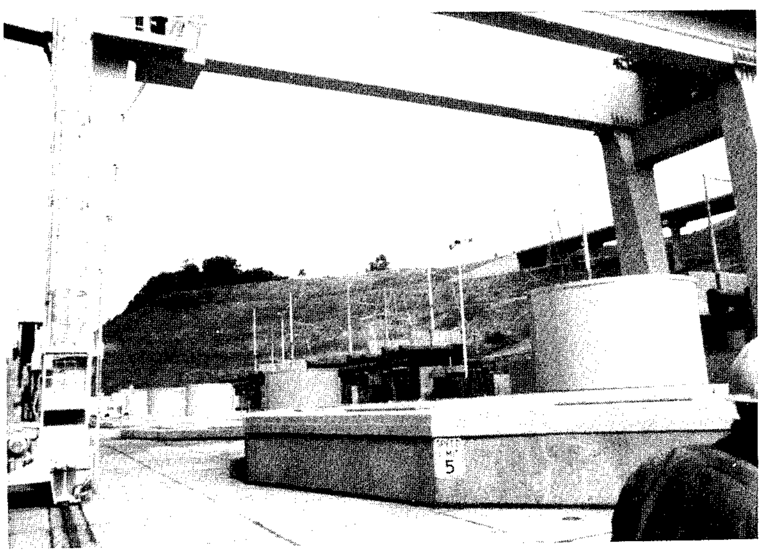


Figure 1

Before disassembly, a complete set of as-found readings were taken. These readings will be very helpful during reassembly and will also help in determining what repairs may be necessary.

The actual disassembly started with the generator top cover. This cover had to be stored on top of the powerhouse and be left pre-rigged at all times. Since this cover is the unit's only weather protection, it had to be ready for reassembly at a moment's notice. Special provisions were made if the weather deemed it necessary; the cover could be successfully mounted in less then 15 minutes.

The generator was completely disassembled except the rotor rim, the stator and the brake ring. The rotor assembly was much too heavy for the crane. Consequently, all 64 rotor poles were removed and the rotor spider removed from the rim. The rotor rim had to be heated after the poles were

removed and before the spider could be removed. Both upper and lower generator bridges were removed as well as both guide bearings and the thrust bearing.

The pump/turbine was disassembled completely except for the embedded parts and the bottom ring. After the disassembly, all components were cleaned and inspected using the various typical types of NDE.

Description of Repair

GENERATOR

After the disassembly and inspections were complete, the repair portion of the contract started. The generator repair consisted of the following items:

1) Approximately (30%) of the stator coils were rewedged.
2) All of the generator bearing oil coolers were cleaned and pressure tested.
3) All valves were either rebuilt or replaced. These included oil supply and cooling water valves used in the generator.
4) The Stator air coolers were cleaned and tested.
5) During the NDE inspection of the rotor spider arms, cracks were found at the hub to spider leg attachment. These cracks were weld repaired.
6) The rotor rim iron and stator iron were inspected and necessary repairs made.
7) All the brake shoes were replaced.
8) The rotor poles were transported to a nearby climate controlled warehouse where the pole rehabilitation took place.
9) The thrust bearing shoes were rebabbitted.

Upon completion of the repairs to the generator, the stator was cleaned and painted. Also, a Partial Discharge Detection system was installed so that plant personnel can monitor the condition of the generator regularly.

PUMP/TURBINE

The pump/turbine repair consisted of the following:

1) Repair of corrosion and cavitation on all accessible surfaces of rotating and non-rotating parts amounted to 225% more than originally anticipated.

Due to the time involved to perform this amount of cavitation repair, it was decided to start the repair on the runner while the disassembly was still in progress. The work platform was installed and a ventilation system was put into place. Armed with the proper safety equipment, the FMS welders went to work.

Vent opening measurements of the six bucket runners were taken before the excavation started and monitored during the whole repair process. All cavitation repairs were done using 309L stainless steel weld wire. Most of the repairs were welded using a semi-automatic wire feed procedure.

The completion of the cavitation repair took place in the erection bay building after disassembly.

2) On site field machining processes:

a) The discharge ring and parking ledge were badly cavitated. Consequently, a machining device was used to machine the cavitation to sound base metal before welding with 309L stainless steel. Upon completion of the welding, the machining device was re-aligned to an established unit centerline and finish machined.

b) The decision was made to reinstall the headcover sections and check the concentricity of the wicket gate bores in the headcover and bottom ring. After the check it was determined to line bore the wicket gate penetrations.

c) As we started to line bore the wicket gate penetrations, additional inspections revealed a problem with the lower bushing counterbores. The inspections indicated the counterbore depths were not consistent with drawing dimensions. Consequently, during the line boring operation we set and machined the counterbores.

d) During the inspection of the wicket gate bore alignment, we also checked the perpendicularity of the thrust liner mounting face on the headcover. These faces ran out as much as (.018"). We, therefore, designed a machining device to machine these surfaces perpendicular as shown in Figure 2.

e) Actual runout on the upper runner rotating wear ring was (.067"). The runout on the lower runner rotating wear ring was (.096"). A machining device was designed and built to machine the runner at site. Since the runner weighs approximately 330 tons, the design of this device was a challenge. The runner was supported by a thrust/guide

bearing assembly capable of supporting 500 tons. The drive assembly used to rotate this assembly was comprised of a 150 horsepower hydrostatic pumping unit which drives two hydraulic motors which in turn rotated the column through a bull and pinion gear arrangement. This assembly is shown in Figure 3.

Each wearing ring was machined concentric to one another and to within .030" to .040" of round. The lack of extra wear ring material prevented us from machining to a perfect round condition.

f) During disassembly, 9 of the 16 rotor spider keyways suffered some amount of damage due to galling. Each of these required grinding, weld build-up and remachining to return them to original condition. The site machining operation is shown in Figure 4.

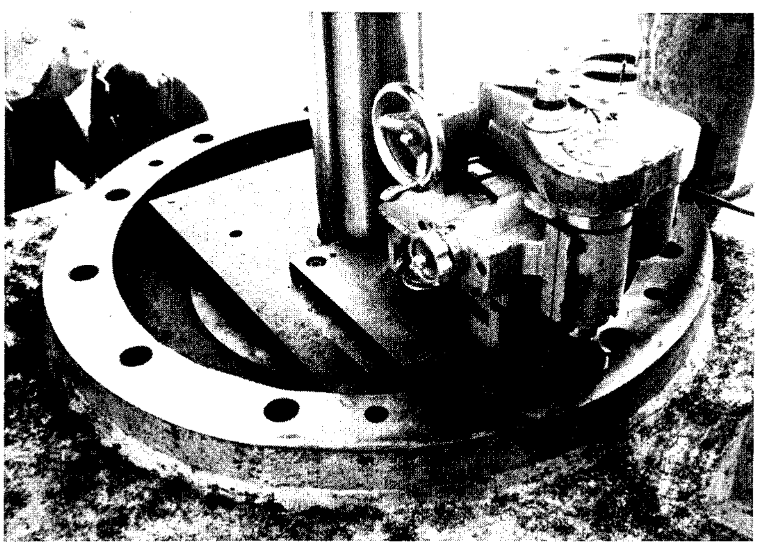
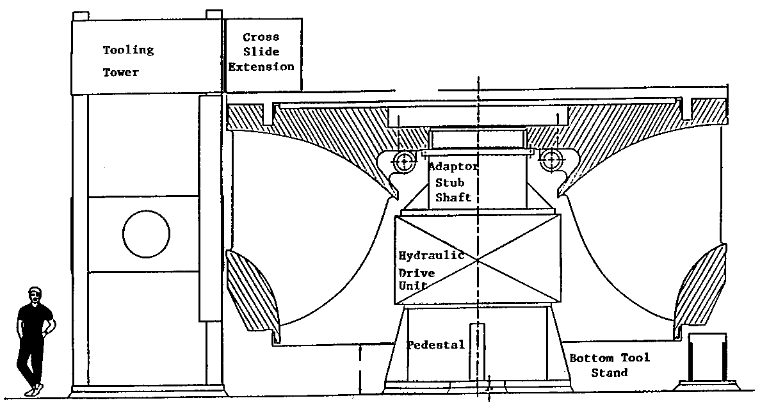
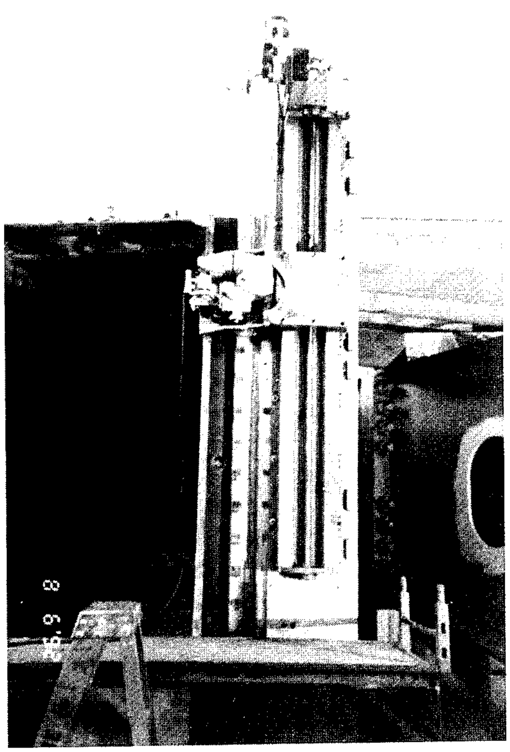


Figure 2



Figures 3 and 4



3) Field Painting

The items to be field painted were the stay vanes, interior of the discharge ring, upper end of the draft tube, top face and bore of the bottom ring, water immersed surfaces of the headcover, and the runner with nose cone.

Originally all of these items were to be painted with either coal tar epoxy or Carbomastic 15. The customer requested an alternate coating which would be a lighter color and easier to illuminate within the water passages. Valspar 78-W-3 met this criteria and used in place of both the coal tar and Carbomastic 15.

Many additional items were added to the base scope of the contract at the customer's request. They are as follows:

a) The Penstock Expansion Joint is in a separate chamber from the powerhouse. The surface area in this chamber that required painting was approximately 3200 square feet.
b) The bottom area of the turbine shaft, (below the packing sleeve) and the new nut guard.
c) Both the upper and lower stay ring crown plates.
d) All of the stay ring pockets which are adjacent to the outside diameter of the headcover.
e) All non-machined surfaces of the headcover sections.
f) All non-machined surfaces of the turbine pit after the reassembly was complete.

4) Replacement parts and shop repair

a) The refurbishment of all three stem journals on the 20 wicket gates was accomplished at Voith Hydro's facility in York, Pennsylvania. The repair procedure was to rough machine each journal surface, weld overlay using type 309L stainless steel filler material, and finish machine to O.E.M. specifications.
b) Replacement of all wicket gate bushings along with associated packing and seals.
c) Replacement of the lower stationary wear ring including drilling, tapping, and reaming for new attaching screws and dowel pins.
d) The wicket gate arms, thrust liners, and the packing box support were remachined in Voith's facility.
e) Completely disassemble both wicket gate servomotors and

replace piston rings, bushings, and packing. The servomotor piston rod and rod eye were replaced in the south servomotor due to damage found during inspection.

f) Replacement of all the gate ring wear pads and wicket gate linkage bushings.

Reassembly

The reassembly of Unit 5 occurred without incident. Some of the repair activities greatly improved the efficiency of the reassembly effort. For example, we avoided the shimming of the thrust liners by correctively machining the headcover seats and the liner bodies.

The thrust bearing shoes were equally loaded using a site designed strain gage system. This system accelerated the schedule and proved to be extremely accurate.

Post-Outage Testing

Our two primary tasks as given by the customer (LPS Plant Management) were to correct the problem of runner wear ring contact, and reduce the leakage of water through the lower gate stem bushings. Both of these goals have been met and substantiated in our post-outage test reports.

As with the pre-outage testing, the post-outage testing was a joint effort between contractor and customer. The post-outage testing also indicated a definite increase in unit efficiency as well as a decrease in unit vibration. Cycle efficiency was improved by over 11%. O.E.M. design clearances were met between the stationary and rotating wear rings. Due to the concentric clearances between stationary and rotating wear rings the runner band pressure was stabilized and, therefore, the hydraulic radial forces equalized. This also reduced the thrust bearing loading under certain operating conditions.

Conclusion

The efforts by all involved to identify and correct problems, and the care taken to reassemble the unit have resulted in a smoother, more efficient running unit.

Despite accomplishing 20% more work then originally planned, the outage was completed ahead of schedule and below budget.

SENECA P/T REHAB.
FIELD TESTS AND ECONOMIC PAYBACK

Ronald E. Deitz *
James L. Kepler **

Abstract

This paper presents the results of the runner upgrade and unit rehabilitation of two pump/turbines at the Seneca Powerhouse. Topics of discussion are the volumetric testing method utilized for field testing Unit 2 and the results of these tests before and after rehabilitation and their economic benefits.

Background

The pump/turbines at the Seneca Pumped Storage Powerhouse were manufactured by Newport News Corporation in 1967. The powerhouse contains 2 - 224" (5.69 m) pump/turbines and one smaller Francis turbine.

The Seneca project is located in the northwest portion of Pennsylvania near the town of Warren.

* Manager, Field Engineering, Voith Hydro, Inc., P.O. Box 712, York, Pennsylvania, 17405, Telephone: (717) 792-7000

** Manager, Service Marketing, Voith Hydro, Inc., P.O. Box 712, York, Pennsylvania, 17405, Telephone: (717) 792-7000

OPERATING CONDITIONS

	GENERATING	PUMPING
Maximum Output	351,000 HP (262 MW)	3,700 CFS (105 m 3/s)
Maximum Head Unit 1	800 FT (255 m)	820 FT (250 m)
Maximum Head Unit 2	851.5 FT (259 m)	820 FT (250 m)
Minimum Head	630 FT (192 m)	660 FT (201 m)

The layout at the Seneca plant is unique as shown in Figure 1. The upper reservoir consists of a manmade captive pond. A single penstock runs from the floor of the upper reservoir to the powerhouse. The powerhouse has two reversible pump/turbines rated at 175MW each and a Francis turbine rated at 30MW. The first pump/turbine is connected to a lower reservoir at Kinzua Dam. The second pump/turbine is not only connected to the same lower reservoir but can also discharge flow in the generation mode to the Allegheny River below Kinzua Dam. All pumping is done from the lower reservoir. The Francis turbine discharges water only to the Allegheny River. Details of the plant can be found in reference 1.

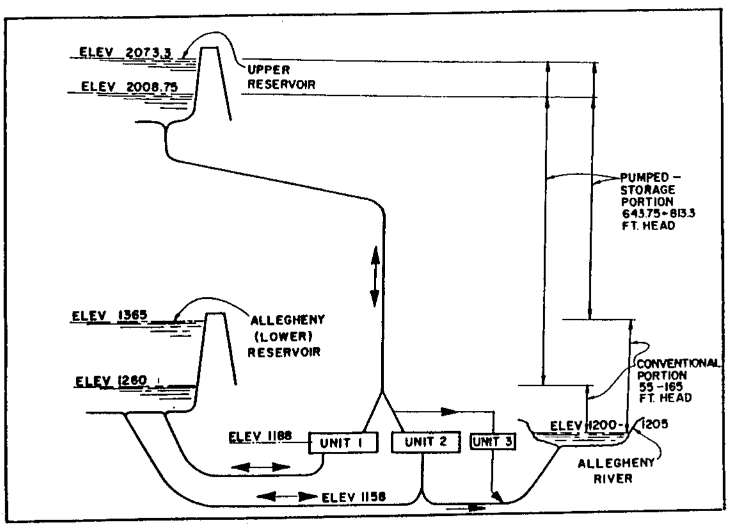


Figure 1
Seneca Project
Water Flow Diagram

Field Test Measurement Methods

Flow rate was measured by the volumetric method utilizing the upper reservoir. The upper reservoir is a man made captive pond which is approximately 750 meters in diameter and 25 meters in depth. The main variables required for flow determination are area verses elevation and the rate of water elevation change. The only sources of water in or out of the upper reservoir are from the penstock, rain, leakage and evaporation. Upper reservoir volume was obtained from surveys taken in 1966. Four equally spaced locations were used to determine the elevation of the upper reservoir. It was not practical to measure the water surface so pressure transducers were placed on the floor of the reservoir.

Effective head on the pump/turbines was difficult to measure because there were no pressure taps located in the draft tube exit. The total energy at the spiral case (high pressure side) was determined from the standard net head taps. The total energy on the draft tube side or low pressure side was obtained by measuring the tail water elevation and correcting for the head loss in the tailrace tunnels. The water surface on the low pressure side was measured utilizing ultrasonic level transducers by Miltronics model ST25.

Motor/generator output in both kW and Vars was measured with a RIS model PCM-30 watt/var transducer utilizing the three-phase four wire method. Exciter power was measured with a Scientific Columbus model 6268 DC watt transducer utilizing direct voltage and shunt resistor for current. Pump and turbine power was calculated based on motor and generator efficiency.

Test Procedure

1. Prior to calibration of any equipment, all transducers were connected and energized with the same cables used for the test. This minimized any possible effect of cable lengths and power supply loads. Calibration was through the data channel (transducer, cable, power supply, filters, and recording system).

2. Transducers were mounted and references established.

3. A static check is very important. This required a short test which measured the transducer outputs with the unit shut down and pressurized. The transducer outputs were verified. Discrepancies were resolved prior to continuation of the test.

4. A reservoir leakage test was performed to determine the influence of leakage and evaporation. This test required the plant to be shut down for 24 hours.

5. Prior to each volumetric test, a piezometer traverse was conducted with the unit at test conditions. This information was used to determine if the measurement location was satisfactory for head measurements. Test codes required individual taps from the average of all taps to be within 20% of the velocity head and opposite pairs to be within 10%.

6. The volumetric test consisted of measuring the change in volume of the upper reservoir with respect to time along with the power and head. To obtain an accuracy within approximately 1% for flow rate, it was necessary to run the test for about three meters change in upper reservoir elevation. Three meters is approximately 1.3% change of head. Total working range of the upper reservoir is approximately fifteen meters which provided for three meter test ranges. For the generation cycle, the desired gate position was set and held constant by the governor gate limit. Best gate was maintained for the pump cycle. For pump tests, the gate position followed the optimum gate to head relationship. The computers were given the starting elevation. When the head water reservoir elevation reached the starting elevation, the program started taking test data. After 3 meters change in upper reservoir elevation, the test point was completed.

Computers and Programs

Each test point requires approximately four hours. Pump/turbine plants normally require testing to be done during daytime hours for the generation cycle and nights for the pump cycle. To minimize the manpower requirements, all test data was obtained by computer. With the equipment and setup, the test was run by a plant operator and one person to operate the computer. A second person was used to assist with piezometer traverses and possible equipment problems.

It was not possible to direct connect the head water transducers to the computer in the powerhouse. It was necessary to use two Toshiba 5200 20MHz 386 computers, one located at the headwater and the other in the powerhouse. Both computers were linked by modems on one of the plant's dedicated phone lines. The testing was performed in the powerhouse. The data acquisition was done with a program written in FORTRAN using Discovery Data Inc. model ADA64 analog to digital

boards. BASIC language was used for the communication program between the two computers. From the start of the test, the following process was performed each minute to acquire the test data analyze the results.

- A data point was taken of all 21 data channels. The data point consisted of an average of 100 data samples plus the standard deviation. The average and standard deviation was stored in data files on the hard disks.

- The data point average and standard deviation of the four head water transducers was transferred from the computer at the head water to the computer in the powerhouse. The water temperature was transferred from the powerhouse to the head water. This temperature was used to determine the water density which was then used to convert pressure to feet of water.

- Afterwards the powerhouse program calculated flowrate, power, net head, and efficiency for each test point and integrated these values from the start of the test. These values along with the transducer readings and the error band based on a 95% confidence level were displayed.

Testing Difficulties

The first difficulty to overcome was the need to tie two computers together. This plant had a spare dedicated telephone line, so the connection was completed by modems.

The next difficulty was to transfer the data from the four upper reservoir transducers to the computer in the powerhouse. In the program development phase, it became apparent that the optimum set-up would be to have each computer run two programs simultaneously with the computer in the powerhouse controlling the testing progress. Each computer would require a program to acquire the test data and a program to transfer data between the two computers. This would permit the test to be salvaged if a communication problem developed during the test. If both data acquisition and data transfer were in the same program, the data acquisition portion would be interrupted if a problem developed during the data transfer (noise on the phone line would easily cause this problem). Any interruption in the data acquisition in the powerhouse would abort the test because of the integration of power and head over the test period. Running two programs was not possible with the DOS operating system and at the time of program development, OS2 was still having problems.

DeskView software was used permitting each computer to simultaneously run a communication program and a program for data acquisition and analysis. During the course of the testing, communications were lost on several occasions and this feature eliminated the need to rerun a test.

The next difficulty was to find a reliable and accurate method of measuring the headwater. Some submersible transducers had been used in the past, but have had problems. The submersible transducers used in the past were differential type transducers with the low pressure side vented to atmosphere by a small capillary tube hidden in the factory wiring. Failed transducers were returned to the factory and all had the same problem, water in the vent tube. The water may have resulted from condensation and not a leak in the vent capillary. To eliminate this problem, waterproof Viatran Model 574V15 differential pressure transducers were used. A large vent tube was connected to the low pressure side and routed externally to the transducer wiring. If water found its way into the transducer, it could easily be corrected and the transducer would still be functional. The transducers were mounted in weighted containers and lowered to the bottom of the reservoir in specific locations.

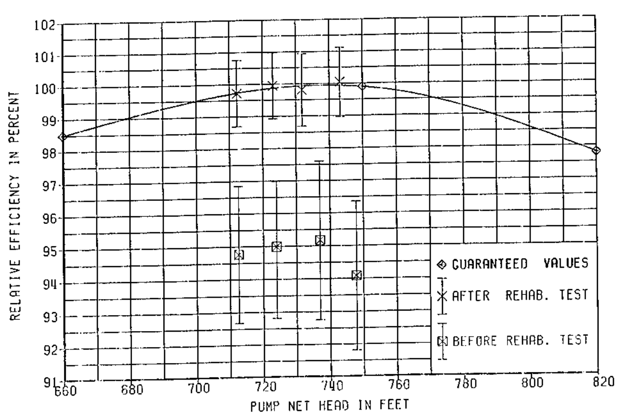


Figure 2 - Comparison of Before and After Rehab. to Guarantees - Pumping

The upper reservoir elevation was measured with 4 to 20 milliamp current loop transmitters because the distance from the transmitter to the computer was up to 1000 meters. All four transducers were powered by a

common power supply for the test before the rehabilitation. The transmitter with the longest length of wire had about twelve times more calibration error than the transmitter with the shortest length of wire. For the test after the rehabilitation, each transmitter had a power supply. This improved the calibration error with the longest wire 2.6 times greater than the shortest. The result of decreasing this error band reduced the overall test inaccuracy by half as can be seen on Figures 2 and 3.

Test Results

Comparison of Unit 2 test results before and after rehabilitation along with the guarantees are shown in Figure 2 for the pumping mode and Figure 3 for the generation mode. There is an average increase in efficiency of 5.4% in the pump mode, 6.1% when generating to the lower reservoir, and 3.9% when generating to the river. Cycle efficiency (defined as energy generated divided by the pump energy) increased by more than 9%.

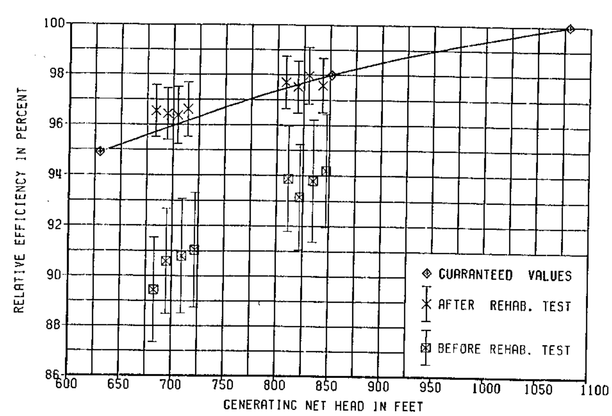


Figure 3 - Comparison of Before and After Rehab. to Guarantees - Generation

There were other improvements gained by the upgrade and rehabilitation. Unit 2 pump/turbine has an unusual draft tube design. This was required so the unit could discharge water either to the lower reservoir or the river. As a result, there was a vortex in the draft tube while pumping, which sounded and acted similar to a Rheingans phenomenon which normally occurs at part load in the generation cycle. This condition

was investigated during the model tests. Based on these tests the modifications to the draft tube eliminated the rough pump operation which should save considerable wear on the unit.

The axial hydraulic thrust was decreased by a factor of five base on thrust bridge deflections. This improvement is attributed to the poor condition of the impeller and seals prior to rehab and the new labyrinth type impeller seal.

Economic Considerations

The production of electric power to satisfy system demands is the ultimate goal and of utmost concern for most power producers. Achieving these goals efficiently and economically are both of equal importance. The Seneca pump/turbine upgrade project attributed to Pennsylvania Electric Company in their efforts to achieving those sound economic goals.

In order to provide Pennsylvania Electric Company with the best performance and efficiency, specific areas of the pump/turbine had to be examined and improved where possible. Those areas were the runner (impeller), runner seals, hydraulic flow surfaces, hydraulic shapes and head and tailwater variations. The improved performance guarantees were checked by model and subsequent field tests.

This paper uses actual final field test data as shown in Figure 4 for Seneca's Unit 2 pump/turbine, to study the economic benefits of the upgrade and rehabilitation program.

For the purpose of this presentation, a simple analysis of direct payback improvements and comparison of present value is used. An average conservative value of $.014/KWH for pump power costs and a conservative average of $.021/KWH for generation sell is assumed. Figure 4 lists actual field test results. For time, we will use 60 hours per week average pumping at 200 MW and 40 hours per week average generating at 200 MW.

Pumping Analysis

For our pumping analysis we will use our improvement of 6.82% (average) as noted in Figure 4 x 60 hours per week x 200 MW x 52 weeks per year x .0682 = 42,600 MWH per year savings. Using the 42,600 MWH x 1000 x $0.014 per KWH = $596,000 cost savings to Pennsylvania Electric Company per year for Unit 2.

ACTUAL SENECA FIELD VOLUMETRIC TEST RESULTS
MOTOR INPUT AND GENERATOR OUTPUT
BEFORE AND AFTER REHABILITATION

(PUMPING OPERATION)

HEADWATER RANGE (FT) ELEVATION		PUMP INPUT POWER OLD (MW HOURS)	PUMP INPUT POWER NEW (MW HOURS)	PERCENT OF IMPROVEMENT	DECREASE IN INPUT OLD MINUS NEW (MW HOURS)
2066.5	2057.6	865.1	801.5	7.94%	63.6
2053.9	2045.75	746.9	691.8	7.97%	55.2
2043.9	2034.1	840.2	793.4	5.89%	46.8
2032.2	2022.5	786.0	745.3	5.46%	40.7

IMPROVEMENT 6.82% (AVG)

(UNIT 2 GENERATION TO RESERVOIR)

HEADWATER RANGE (FT) ELEVATION		GENERATOR OUTPUT POWER OLD (MW HOURS)	GENERATOR OUTPUT POWER NEW (MW HOURS)	PERCENT OF IMPROVEMENT	INCREASE IN OUTPUT NEW MINUS OLD (MW HOURS)	CYCLE EFF. [GEN/PUMP] OLD (PERCENT)	CYCLE EFF. [GEN/PUMP] NEW (PERCENT)
2066.5	2057.6	569.7	602.3	5.72%	32.6	65.9	75.2
2053.9	2045.75	489.2	517.6	5.79%	28.4	65.5	74.8
2043.9	2034.1	550.9	591.6	7.38%	40.7	65.6	74.6
2032.2	2022.5	507.9	553.0	8.89%	45.2	64.6	74.2
		IMPROVEMENT 6.95% (AVG)				65.4	74.7

9.4% AVG IMPROVEMENT IN CYCLE EFF.

(UNIT 2 GENERATION TO RIVER)

HEADWATER RANGE (FT) ELEVATION		GENERATOR OUTPUT POWER OLD (MW HOURS)	GENERATOR OUTPUT POWER NEW (MW HOURS)	PERCENT OF IMPROVEMENT	INCREASE IN OUTPUT NEW MINUS OLD (MW HOURS)	CYCLE EFF. [GEN/PUMP] OLD (PERCENT)	CYCLE EFF. [GEN/PUMP] NEW (PERCENT)
2066.5	2057.8	696.1	720.0	9.67%	23.9	80.5	89.9
2053.9	2045.75	596.6	624.3	9.12%	27.7	79.9	90.2
2043.9	2034.1	674.4	709.1	9.51%	34.7	80.3	89.4
2032.2	2022.5	642.9	665.0	9.67%	22.1	81.8	89.2
		IMPROVEMENT 9.49% (AVG)				80.6	89.7

9.1% AVG IMPROVEMENT IN CYCLE EFF.

Figure 4

Generating Analysis

For our generation analysis we will use our improvement of 9.49% (average) to the reservoir and 6.95% (average) to the river as noted in Figure 4. This combined average will then be .0822 x 40 hours per week x 200 MW x 52 weeks per year = 34,200 MWH per year increase in generation. Using the 34,200 MWH x 1000 x $0.021 per KWH = $718,000 cost improvement to Pennsylvania Electric Company per year for Unit 2.

In addition, an estimated annual maintenance cost savings of $200,000 per unit will be recognized, which provides additional benefit for performing the upgrade and unit rehabilitation.

Summarizing all positive effects, the benefits from pump power reduction of $596,000, plus generation revenue increases of $718,000, plus maintenance cost savings of $200,000, we can recognize a total cost benefit to Pennsylvania Electric Company of $1,514,000 per year for their unit number 2 upgrade.

Present Worth

Calculating the present worth over 20 years, at an interest rate of 8%, adding the annual pump savings of $596,000, plus the annual generation improvement of $718,000 and $200,000 annual maintenance savings the present worth is $14,900,000 to the utility for Unit 2.

Conclusion

The economic studies and field tests conducted for Unit 2 pump/turbine, truly justify an economic payback in excess of 2:1, "benefit to cost ratio" for undertaking the upgrade and rehabilitation program of the pump/turbines at the Seneca plant of Pennsylvania Electric Company.

References

Rock, Seymour B., Senior Engineer, "The Seneca Pumped-Storage Plant", Proceedings of the American Power Conference, Volume XXIX, 1967.

Franke, Gary F. and Kepler, James L.; "Case Study in the Upgrade and Rehabilitation Techniques of a Pumped Storage Installation", Waterpower '91, Volume 3, Civil Works, Construction, Hydraulics, Mechanical Sys., ASCE, 1991.

YARDS CREEK PUMP/TURBINE UPGRADE
Part 1 - Structural Redesign and Analysis

J.R. Degnan[1] J.J. Geuther[2]

ABSTRACT

Yards Creek pumped storage station located in northwestern New Jersey has three identical 120 MW Francis type reversible pump-turbines. These machines were commissioned in 1965 and have since experienced serious fatigue damage and fracture in their major components, including the original runner, headcover, stayring and some wicket gate linkage elements. For example, stayvanes have fractured up to 1.8 m in length on several occasions. One instance resulted in a secondary stayring fracture, which under 213 m of head, flooded the pump/turbine pit. Another annoying problem involves the 3rd harmonic of the runner vane's passing frequency exciting the fundamental wicket gate torsional vibration mode. This results in rapid wear of the gate's upper guide bushing and associated shear pin fatigue. With the open bushing clearances, the operators are plagued with multiple shear pin failures every week! The early runner and headcover problems and their solutions are well documented in previous literature. [1,2] This paper describes the stayvane and wicket gate modifications in light of their reasons for failure as verified by structural analyses and field testing. While the modified design promises significant life extension of the currently distressed machine elements, the modified stayvane and wicket gate shapes also help to enhance the machine's hydraulic performance in concert with the new runner design described in Part II of this series.

STAYRING

The last scheduled overhaul of the pump-turbines was done in 1981 to replace the original pump-turbine headcovers and to repair wicket gate cavitation damage. Repair and reinforcement of the stayring bolting flanges of Units 1 & 2 were also done concurrent with this work. More recently the stayrings of all three units have experienced cracks in several of the

[1]V.P. Engineering, American Hydro Corporation, York, PA 17402-3039.
[2]Senior Engineer - Generation, Jersey Central Power & Light Company, Yards Creek Station, Blairstown, NJ 07825.

stayvanes. Most of the cracks started from small defects near the pump entrance edge and propagated outward along the lower fillet toe of the vanes. Weld repairs of these stayvanes have all been successful, but the probability of serious brittle fracture increases as fatigue damage accrues. The stayring consists of A148 90 - 60 cast steel, a material which, in spite of its high strength, is quite intolerant of casting flaws commonly found in areas of section transition such as the failure sites. The fracture toughness of the material was tested at minimum service temperature to be only 42.9 MPa $\sqrt{m}$ {39 KSI $\sqrt{in}$} (correlates to about 6.8 J {5 ft-lbs} charpy impact energy) [3]. For a given maximum tensile stress, fracture toughness determines the critical crack size beyond which stable crack growth per load cycle ceases and theoretical fracture occurs. Its value is strongly influenced by the material's Fracture Appearance Transition Temperature ($FATT_{50}$)[1] which for A148 90-60 is 52°C, well above the service temperature range. This inadequate toughness of the stayring, which experiences several daily local stress changes up to the material's yield point, is the cause of fracture in both the stayvanes and upper crown ribs. The following diagnostic analyses of stress, deformation, fatigue and fracture life clearly lead to this conclusion and provide the understanding needed to devise a repair of the structure which promises reliable life extension.

Failure Analyses - Figure 1 shows a mathematical model of the stationary pump/turbine components assembled with high order cubic subparametric finite elements. This axisymmetric computer model was used to assess the interactive loading on the stayring by the headcover, discharge ring and spiral case components for four significant operating conditions:

1. Generating maximum power (steady state)
2. Pump prime
3. Waterhammer due to rapid gate closure
4. 3 unit load rejection

Table 1 gives the magnitude of the reported interaction forces which were used as boundary conditions on a three dimensional mathematical model of the stayring structure shown in Figure 2a. This model provides three orthogonal displacements per node and all their first derivatives which are the direct and shear strains needed to compute the state of local surface stresses throughout the structure. Such accurate identification of maximum stresses and their location are required to properly evaluate fatigue life and determine the integrity of this stayring. Figure 3 shows that the stayring is quite flexible when compared to modern pump turbine designs as evidenced by the non-uniform circumferential distribution of headcover uplift into the upper crown flange. This flexibility prevents the headcover uplift and spiral case membrane loads from distributing evenly into the stayvane's

1. $FATT_{50}$ is the temperature below which a ductile material exhibits brittle fracture characteristics [4].

cross sectional area, causing the damaging local stresses shown in Figure 4 for pump prime condition. Note the good correlation between analytical and field measured stress results [5]. Pump prime is the most aggressive daily loading accounting for 80 percent of the crack growth leading to fracture. Analyses show that cracking to date did not initiate by fatigue but grew from tiny, pre-existing casting flaws. Actually, fracture mechanics analyses of an edge crack at the failure site warns of a brittle fracture once the crack grows beyond 2.8 mm (.11 inch) in length! The initial size of this casting defect in 1965 would have been too small to detect even using today's non-destructive examination technology. Scanning electron microscopic examination of an early fractured vane surface, in fact, showed the presence of such a tiny casting flaw from which a 1.8 m long brittle fracture occurred.

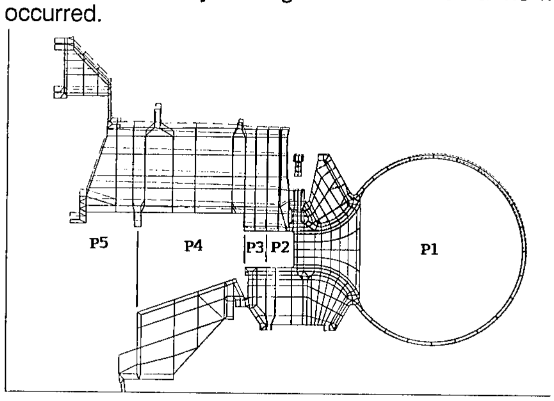


Figure 1 - Yards Creek Existing Distributor Analysis - Pump Prime

TABLE I
OPERATING PRESSURES, kPa AND INTERCOMPONENT LOADS, kN

Location in Figure 1	Normal Generation	Pump Prime	Water-hammer	3-Unit - Load Rejection
P1	2068	2585	3500	3500
P2	1655	2585	3500	3500
P3	1470	2760	160	3500
P4 (Parabolic)	1036-->1470	2326-->2760	160	1775-->3500
P5	160	345	160	160
Headcover Uplift	36,222	68,512	18,142	74,432
Maximum Spiral Case Pull	64,785	80,981	109,702	109,702
Nominal Axial Vane Stress, kPa	66,700	95,685	90,085	121,410
Headcover Radial Pull	19,572	31,138	18,238	36,920
Discharge Ring Radial Pull	7,117	10,231	9,786	13,345

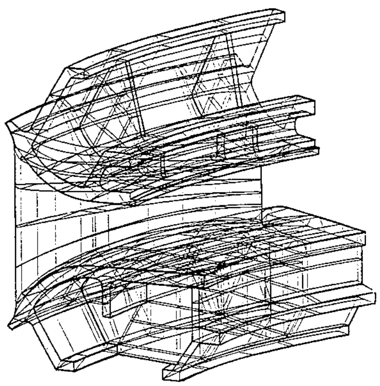


Figure 2a - Existing Stayring

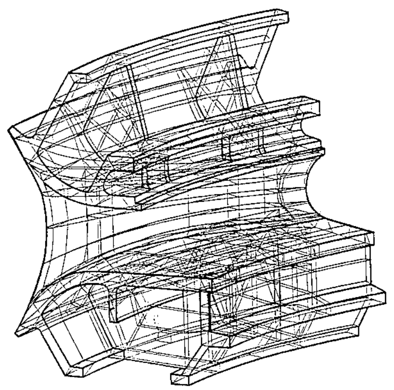


Figure 2b - Modified Stayring

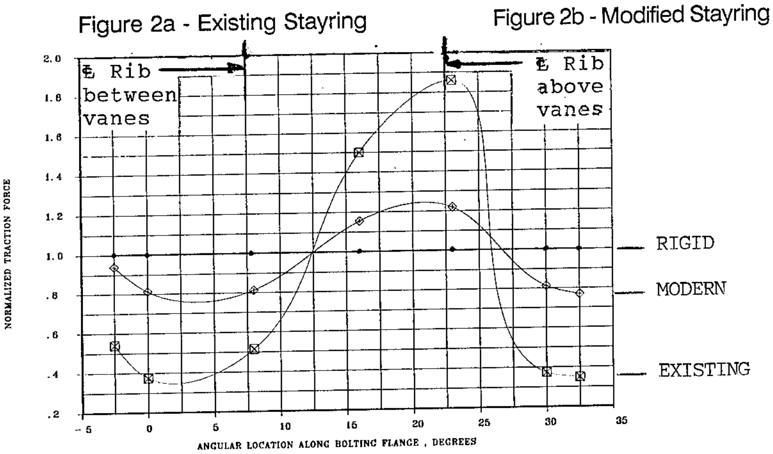
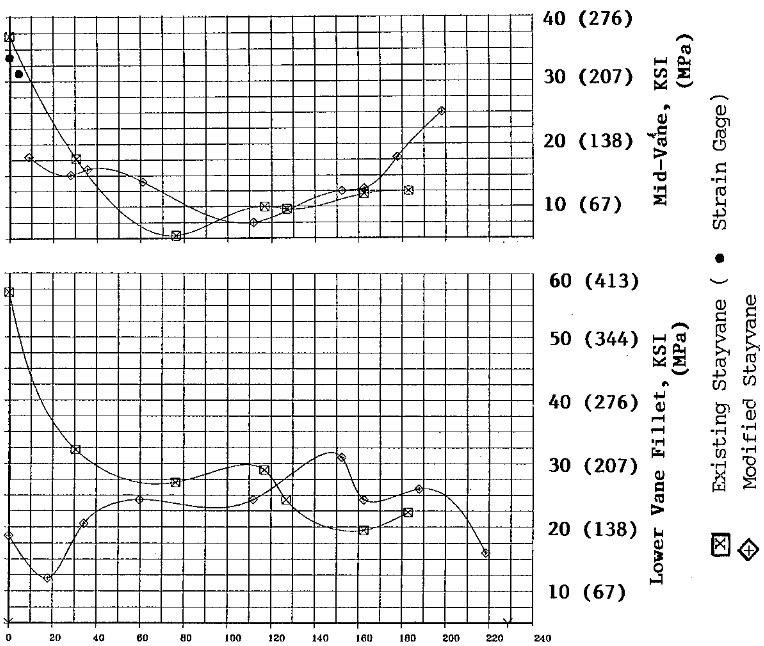


Figure 3 - Headcover Uplift Distribution for Level Stayring Flange Interface

Modified Stayvane Design - The same analytical tools used to correctly diagnose the cause for fracture of the existing stayvanes were used to develop and verify the repair design. Evaluation of different concepts showed that a near optimum design could be achieved using most of the existing structure, which is imbedded in concrete five stories underground. Only the highest stressed inside regions of the 12 stayvanes needed replacement with very ductile 304 stainless steel plate inserts as shown in Figure 5. This material has four times the fracture toughness of A148 90-60 steel at 0°C and boasts a $FATT_{50}$ <-200°C. Note the "scalloped" vane's leading and trailing edges. This feature redirects the headcover uplift and spiral case pull through the thicker mid-section of the stayvanes. In fact, local stresses along the inner edge are now optimized at a near constant

level, having been reduced to 1/3 of the original maximum stress. Figure 5 also compares the modified vane design with the existing vane profile. Note the tapered thickening of the lower 25 percent of the modified vane tip to control the stress riser associated with the stiff imbedded bottom ring support. Figure 2b presents the mathematical model used to verify the redesigned stayring including the 12 additional upper crown tangential ribs discussed below. Figure 4 shows the dramatic reduction in maximum local vane stresses relative to those in the existing vanes. Figure 6 presents the improved fracture life afforded by the modified design and selection of 304 stainless steel. The fatigue and fracture life analyses account for the near yield mean stress remaining in the structure subsequent to the non-stress relieved state of the field welded repair . Detailed flaw acceptance criteria have been developed from these analyses in order to assure the intended safety margin for reliable life extension is achieved by the field rehabilitation.



Distance Along Stayvane From Pump Entrance Edge, cm.

Figure 4 - Stayvane Maximum Stresses (Spiral Case Side) - Pump Prime

Upper Crown Radial Rib Reinforcement - Heavy tangential ribs were designed to reinforce 12 of the 24 radial ribs which are located over each of the existing stayvanes. As analytically predicted [3], these highly loaded radial ribs have experienced early extensive cracking due to yield level local

stresses. These stresses developed because of the ribs being in the load path between the headcover uplift above and the stayvane reaction below. Each of these 12 ribs were tangentially reinforced with 70 mm thick x 250 mm wide high strength HY-80 plate steel ribs in 1984 with no incidence of cracking since. During the current rehabilitation of Unit 2, one of the 12 unreinforced radial crown ribs located between two stayvanes was found cracked. Low stresses exist in these ribs which have no direct path to the stayvane reaction beneath. It is likely that this particular rib was flawed or received disproportionate headcover loading. In any case, the crack is being repaired and the 12 remaining radial ribs are being reinforced with 38 mm thick by 200 mm wide tangential ribs of 304 stainless steel plate. This precautionary measure is being taken to assure that any future cracks which may develop will be restrained from cyclic opening thus minimizing their ability to propagate into the primary load carrying members, including the bolting flange or the adjacent cone.

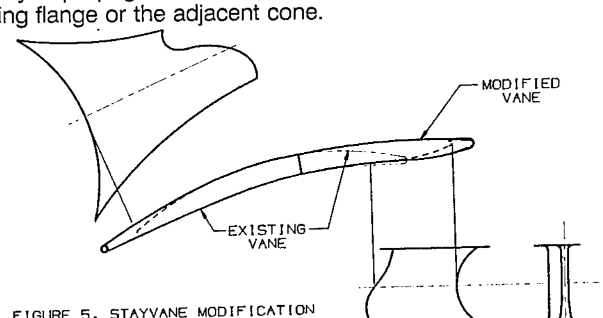


FIGURE 5. STAYVANE MODIFICATION

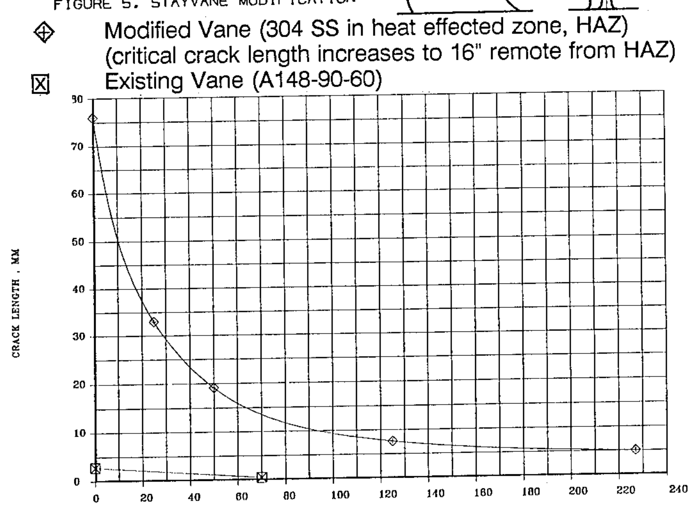


Figure 6 - Yards Creek Stayvane Inner Edge Crack at Lower Fillet Toe

WICKET GATES

Since commissioning, damaging vibration has resulted in the need for periodic refurbishment of the wicket gate assemblies. A program of testing and analyses was conducted in 1987 to diagnose and resolve this problem. It was concluded that resonant vibration could be eliminated by removing a 4-inch-long section of the wicket gate leaf nose, thus shifting the wicket gate's natural frequency. In this case, the project owners needed demonstration that adverse effects on efficiency and wicket gate operation would not result from the modified leaf nose. It was at this time that the model testing presented in Part II of this series was first contemplated.

Field Testing [6] - Late in 1987, an extensive field test program provided important synchronous data relating the existing gates' static and dynamic behavior to their angular opening, the pit noise level, shaft speed and the pressures in the servomotors, spiral case and distributor. This data was collected from start-up through both modes of operation to shut-down. Units 1 and 3 underwent the same test program. Three gates in each machine were selected and instrumented to assess the upper stems'bending and torsional shear strains, the axial strain in each of their links and the three orthogonal accelerations at the top of each stem. Fourier transform analyses were used to reduce this data and determine the frequency response spectrum. Generally, the gate acceleration, stem torque and link loading, all showed predominant response from the first four and up to nine harmonics of the runner's blade passing frequency (240 rpm x 7 blades --> 28 HZ). Figure 7 shows a typical maximum response measured by strain gages on the stem and links during pump start. The accumulation of strain energy is indicative of resonance response to the third harmonic of the runner blades passing the instrumented wicket gates. The measured dynamic shear strains in the stem during pump and turbine start-up and shut-down transients are high with amplitudes corresponding to as much as 50 percent of full squeeze torque. This high cyclic loading verifies the presence of extreme torsional vibration audibly sensed at 125 db in the pit during both transients. Though only about half of the transient dynamic strain is reported during steady state pumping and generating, ambient noise levels remain high at around 110 db.

It is interesting to note that the most severe gate vibration occurs when gate opening is small and the average static gate torque is near zero as measured by differential pressure across the servomotor pistons. Under these two conditions, the existing gates are susceptible to auto-oscillation. This phenomenon may occur at Yards Creek because the unusually long, narrow passageway formed by two adjacent gates speeds up flow as the gates close, causing lower pressures, and slows down flow as they open, causing higher pressures. This results in instability since the gate's instantaneous position does not satisfy equilibrium. This is especially damaging since the resulting vibration is amplified by the resonance condition found by field test and as explained next, by structural analysis.

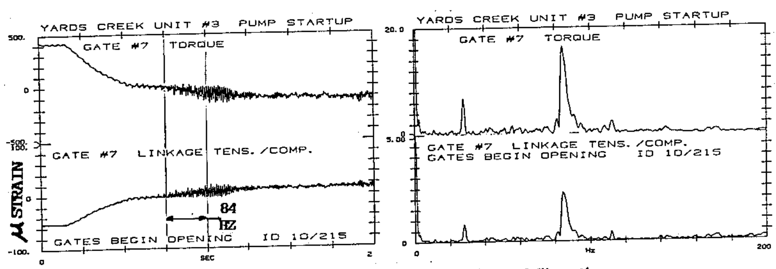


Figure 7 - Measured Wicket Gate Vibration

Vibration Analysis and Gate Modifications - Figure 8 shows the mathematical model and reported fundamental vibration modes of the existing wicket gates supported in their three guide bearings and torsionally constrained with the two - 12,000,000 lb./in. link stiffnesses. Based on the gate rap tests of the gates in and out of water, mode shapes involving leaf interaction with the surrounding water require a 30 percent reduction of their natural frequency when computed for a vacuum. Consistent with the test results, the computed first natural frequency matches the third harmonic of the 28 HZ forcing frequency, thereby exciting the torsional mode. Additionally, a second computed natural frequency matches the fourth harmonic, thereby exciting the upper stem bending mode.

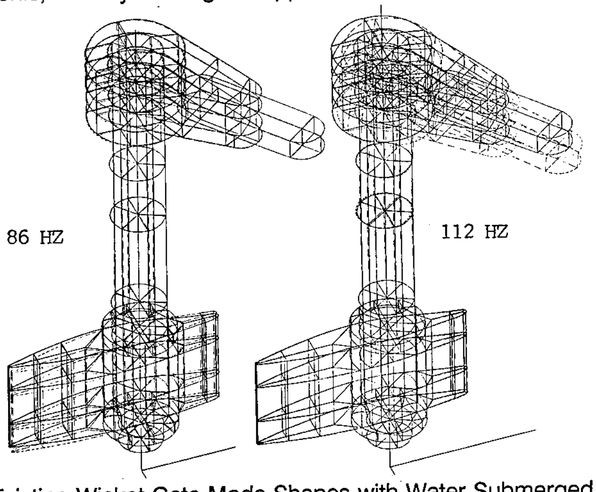


Figure 8 - Existing Wicket Gate Mode Shapes with Water Submerged Leaf

Modified link stiffnesses were considered in order to shift the torsional natural frequency to a mid-harmonic position. However, this easy approach is impractical due to the magnitude of change needed and it ignores the auto-oscillation tendencies of the existing leaf shape at small gate openings.

Figure 9 compares the existing leaf shape with the modified shape whose rotational inertia shifts the gate's torsional natural frequency in water up to 101 HZ, about mid-way between the third and fourth harmonic of the 28 HZ forcing frequency. Also, the narrow intergate waterpassageway has been considerably shortened minimizing the existing gate's tendency to auto-oscillate. In fact, recent comparative model tests have verified up to a 65 percent reduction in dynamic torque for the modified gate leaf. Finally, a shift in the "zero torque" gate position away from speed-no-load operation provides additional protection against vibration. Also, in concert with increasing the gate opening to optimize hydraulic performance, the gate linkage was redesigned. This provided the opportunity to increase the gate arm mass by 113 kg, which was sufficient to shift the natural stem bending mode frequency down to 96 HZ, again midway between the third and fourth harmonic of the 28 HZ excitation frequency. Table II summarizes the improved safety margin against resonance afforded by the modified leaf shape and heavier linkage.

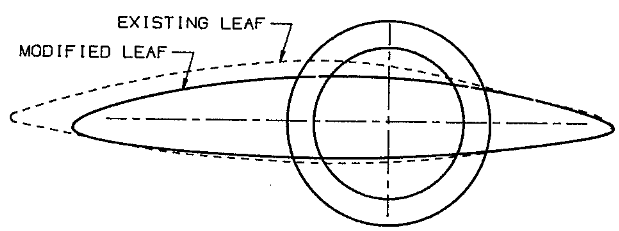


Figure 9 - Comparison of Existing and Modified Wicket Gate Leaf Station

TABLE II - WICKET GATE NATURAL FREQUENCIES, HZ

	EXISTING			MODIFIED NOSE			
Mode #	Vacuum	H_2O	Ratio*	Vacuum	H_2O	Ratio*	Mode Shape
1	112.5	112	4.00	96.2	96	3.4	Stem Bending
2	126.1	86	3.07	148.4	101	3.6	Leaf Torsion
3	176.1	150	5.36	180.7	152	5.4	Combined
4	330.9	330	11.79	261.6	262	9.4	Gate Arm Bend
5	377.0	377	13.47	311.0	311	11.1	Gate Arm Bend
6	553.7	378	13.50	520.2	354	12.6	Gate Leaf Bend

1) {*} Ratio of natural frequency to 28 HZ excitation frequency.
2) Stayvane Von Karman vortex shedding frequencies range from 7.8 HZ at speed no load to 77.1 HZ at full load.
3) Wicket Gate Von Karman shedding frequencies range from 121.3 HZ to 156.7 HZ (turbine direction) and 125.9 HZ to 341.0 HZ (pumping direction) depending on flow rate.

CONCLUSION

In the light of extensive field testing and diagnostic structural analyses, the proposed stayring modification has been designed to remove all fatigue worn vane material and to replace it with tough, crack resistant stainless steel. Stayvane entrance and discharge edges have both been scalloped from top to bottom rings in order to minimize the local stresses responsible for crack formation and the unacceptable intolerance for even tiny material defects in the existing vane. Thus, the safety margin against fatigue damage has nearly tripled and material flaw acceptance criteria has been relaxed to easily detectable flaw sizes. The upper crown radial ribbing has been substantially reinforced with tangential ribbing. These new ribs will reduce damaging local cyclic stresses and prevent residual material flaws from growing into adjacent primary load carrying members. The modified stayvane shape will provide years of reliable machine operation while significantly helping to enhance the hydraulic performance of the pump turbine as demonstrated by recent model testing.

Following extensive field testing and diagnostic analyses, forced torsional wicket gate resonance was found to be the cause of excessive noise, accelerated guide bearing wear, and high cycle fatigue of gate linkage elements. The modified wicket gate leaf has been analyzed for stress, deflection, fatigue life and vibration characteristics. Results indicate that a long life of trouble free operation will follow the modification of the existing wicket gates since the modified leaf shape shifts the natural frequency away from the exciting frequency responsible for the current torsional vibration. It also minimizes auto-oscillation tendencies which existed at small gate openings. Recent model testing, which included the modified gate and stayvane shapes and new runner, showed a 50 to 65 percent reduction in dynamic torque, the driving force exciting the current vibration. Potential resonant excitation of the upper gate stems' bending mode was also eliminated by appropriately altering the linkage mass as part of the redesign for increased gate opening.

REFERENCES

1. Kunich, F., "Runner Headcover and Wicket Gate Experiences - Yards Creek Station", Jersey Central Power & Light Company, Sept. 1982.
2. "Yards Creek Pump Turbine Headcover Analysis - Final Report, Allis Chalmers Corporation, July 1979.
3. "Yards Creek Modified Unit #1 Stayring/Vane Failure Analysis", Allis Chalmers Corporation, May 16, 1985.
4. Barsom J., Rolfe S., "Fracture and Fatigue Control in Structures", Prentice Hall, 1977.
5. "Evaluation of Stayring Service Life, Yards Creek Pumped Storage Station, JCP&L", Stone and Webster Corporation, March, 1990.
6. "Yards Creek Pump Turbine Wicket Gate Vibration Evaluation", American Hydro Corporation, February 5, 1988.

Upgrading of AEP's Twin Branch Hydroelectric Plant

Robert E. Dool[1]
Stefan M. Abelin[2]

ABSTRACT

Until recently, the Twin Branch Hydroelectric Plant (constructed in 1904) had the oldest generating units on the American Electric Power System. By 1984 the plant was beginning to show its age. Unit 1, which was installed in 1922 was out of service due to a turbine shaft failure. Unit 6, installed in 1904, was curtailed to 40% when a similar failure disabled 3 of 5 runners.

American Electric Power began investigations to upgrade the site in 1984. At that time several criteria were established to evaluate the merits of various types of equipment. These criteria were developed to achieve a design which optimized utilization of the river while controlling project costs and environmental impacts.

The criteria included the following:

1) The design should be oriented to keep structural modifications to a minimum and should be compatible with existing foundations.

2) The design should favor installation procedures which minimize the size of, or eliminate the need for cofferdams.

---

[1]Mechanical Engineer, American Electric Power Service Corporation, Columbus, Ohio 43215, (614) 223-2903.

[2]Engineering & Marketing Manager, ITT Flygt Corp., Trumbull, Connecticut, 06856, (203) 380-4700.

3) The design should utilize standardized components while maximizing efficiency to the extent practical.

4) The design should provide for easy access and maintenance procedures.

5) The units need to provide an appropriate level of regulation and operations flexibility.

This paper presents the results of this redevelopment including the economic and operational decisions which lead to the utilization of ITT Flygt submersible turbine-generators.

GENERAL

The original turbines installed at the Twin Branch Hydroelectric Project consisted of multiple Francis runners on horizontal shafts. This design is frequently referred to as a "Camelback" turbine due to the shape of the discharge tubes.

The plant consisted of six units with four to five runners per unit, installed in six open flume intake bays (See Figure 1). The powerhouse is constructed of reinforced concrete on a wood pile foundation.

DESIGN CRITERIA

To evaluate the variety of equipment options proposed for the plant, the project engineers established a set of evaluation criteria based on the major construction components involved.

1. Minimize structural modification

2. Minimize need for a coffer dam

3. Optimize use of standard components

4. Simplify maintenance procedure and access

5. Optimize unit operation control

By combining the cost of the turbine-generators and costs associated with each of the five design criteria an appropriate conceptual estimate could be developed. Further, items 4 and 5 were used to evaluate maintenance support requirements, operational flexibility and life cycle costs.

TURBINE OPTIONS

At AEP's request several vendors provided preliminary proposals, utilizing a variety of unit configurations including Pit Turbines, Bulb Turbines, "S" Turbines, and Submersible Turbine Generators.

A review of the proposal information indicated that all of the designs were plausible and could be adapted to the existing foundations. However, some of the designs required the installation of cofferdams and several would have required significant changes to the powerhouse structures.

The proposals allowed for a variety of control options including both "single" and "double" regulated turbines. In addition, we evaluated the proposed submersible turbine/generators with a simple "on-off" control scheme. Because the flow would be passed in steps of approximately 400 CFS it was necessary to compare the control error to its effects on river flow and headwater elevation. Based on the maximum control error of 200 CFS it was found that the effects to river flow and headwater elevation were within an acceptable range when compared to regulatory limits. Based on current information, we expect to be able to maintain headwater within ± 0.25 ft. Final minimum flow requirements for the plant have not yet been set; however, we believe necessary control can be maintained utilizing the units and spillway gates as necessary at a minimum of cost. Based on this assessment, a step function flow control was judged to be acceptable.

EVALUATION

During the evaluation of the various options several points became clear:

1) The construction of the cofferdams required by some designs is a critical cost factor to this project. Since the stability of the existing structure was not in question, proposals that required cofferdams were at a disadvantage.

2) Since the stability of the structure was not in question, proposals that required additions or changes to the powerhouse were of limited value.

3) For the production levels associated with the plant the additional benefits of "double" regulated equipment could not be justified. It was found that "single" regulated or "on-off" controls provided the best return on the investment.

RESULTS

Based on the available information, it was decided that a plant design utilizing smaller, pre-assembled, submersible turbine-generators would be the most cost effective design for the plant. Several of the advantages have been previously detailed, however it was found during the redevelopment that the design was also adaptable in ways that were not anticipated.

First, since the unit could be installed within the existing intake bays, it was possible to address the replacement of the failed units 1 & 6 in one phase and to complete the replacement of units 2 through 5 at a later date. By doing so, we reduced overall project costs by deferring a significant portion of the capital investment until particular unit performance dictated.

Second, although the six intake bays could accept 2 submersible units each, for a total of 12, only 8 units have been installed (2 each in bays 1 & 6 and 1 each in bays 2-5). This installation was based on the optimum plant economics from current financial models. However, the powerhouse and electrical equipment have been designed to accommodate up to four additional units if future demand makes the addition economically feasible. AEP will periodically review this information and add units as economically justified.

TURBINE SUPPLY

The bid package for the turbine equipment was issued in 1987. After a review of the proposals a contract was awarded to ITT Flygt Corporation for the supply of 4 units with an option to supply additional units as required. The option to purchase 4 additional units was exercised in 1991. The scope of supply included the turbine generators, turbine seats, conical draft tubes, cylinder gates, hydraulic gate activators, accessories and spare parts.

Because of the original bid requirement for a minimum of 1200 kW per bay, Flygt had to design and build their system to accommodate larger generators than Flygt was then using. Each redeveloped bay was to be equipped with two Flygt hydroturbines, each rated at 600 kW. The new generators are 6-pole, 4160V, 600 kW, induction types. They were designed and built during 1988. Assembly and testing were completed in early '89. The turbine is Model EL-7650 with a planetary speed increaser to match the generator RPM.

The concept of installing two turbines, one after the other in each bay was thoroughly tested in model scale, at the Royal Institute of Technology in Stockholm, Sweden. After having fitted flow deflector plates behind each cylinder gate, the test effectively demonstrated the good hydraulic conditions resulting from a well-engineered design. There was no need to modify the size or shape of the existing concrete flumes. The main advantages utilizing the Flygt submersible hydroturbine generators are the simplicity and ease of installation, compact and efficient design, low maintenance requirements, and ease of maintenance. The installation of all associated steel work (draft tubes, seats and gates) could be accomplished from above the flume, which gave a tremendous cost advantage since no excavation or other work was required below it.

CONCLUSION

The evaluation of the alternatives for the redevelopment utilizing critical construction and operational criteria were key to the success of the project.

The plant has been in service with new equipment since 1989 and initial generation figures appear favorable. The estimated annual generation for the completed plant is 33,000 MWh, an increase of 61% over the historical average of 20,482 (1974-1988). This is despite a decrease in the nameplate rating of the plant from 7.26 MW to 4.8 MW or 33%. AEP continues to monitor plant performance.

HYDRO TASK FORCE EVALUATIONS
NORTHERN STATES POWER COMPANY

Roger B. Anderson[1], M. ASCE,
Richard M. Rudolph[2], John R. Larson[3] M, ASCE,
John E. Quist[4] M. ASCE.

ABSTRACT

As part of the relicensing process, Northern States Power Company (Minnesota and Wisconsin) has undertaken comprehensive evaluations of their hydro projects. These evaluations have investigated the condition of the civil, electrical, and hydro-turbine components, estimated costs to maintain and upgrade these components, and set a tentative schedule for repairs through the life of the existing and new license. Northern States Power Company (NSP) formed a team of consultants to evaluate each of these areas, and to identify the recommended repairs and possible upgrades. Nine facilities were evaluated in Wisconsin and one was evaluated in Minnesota. The projects varied in age, complexity, size, and condition; however, the format of the project remained essentially constant for each site.

For each facility, results of the evaluations were compiled into a comprehensive document covering civil, electrical, and hydro-turbine elements. The report provided a tool for planning construction and other site activities. Information developed is valuable for project operation as well as for relicensing. This information provided a basis for monitoring project costs and served as a frame work for short term planning. This paper provides an overview of the entire evaluation process, but focuses on the St. Anthony Falls upper dam site in Minneapolis, Minnesota. Additional evaluations, such as historical and recreational, were performed; however, those evaluations are not presented here.

---

[1]Roger B. Anderson, M. ASCE, Project Manager, Special Projects, Northern States Power Company, 414 Nicollet Mall, Minneapolis, Minnesota 55402

[2]Richard M. Rudolph, Administrator Hydro Engineering, Northern States Power Company, 100 North Barstow Street, Eau Claire, Wisconsin 54702

[3]John R. Larson, M. ASCE, Vice President, Barr Engineering Company, 8300 Norman Center Drive, Bloomington, Minnesota 55437

[4]John E. Quist, M. ASCE, Project Manager, Barr Engineering Co., 8300 Norman Center Drive, Bloomington, Minnesota 55437

INTRODUCTION

Relicensing of hydro projects requires an owner to carefully evaluate a variety of elements, including: 1) component conditions, 2) operations, 3) license imposed restrictions, 4) cost of operations, and 5) possible power upgrade. For most sites, this is a complicated and detailed process. NSP, as part of their relicensing process, formulated a technical team to evaluate many of their hydro projects. Ten plants (nine in Wisconsin and one in Minnesota) were evaluated by a team consisting of Barr Engineering Co. (civil), L & S Electric Corp. (electric), and American Hydro Corporation (hydro-turbine). The objective of each evaluation was to complete a comprehensive review and evaluation of the facility, prepare a report that presents the results, develop cost estimates and possible schedules for completing the recommended repairs. The reports were to be used to evaluate the viability of the facilities and to aid in planning activities.

Completion of each plant assessment provided a comprehensive evaluation of primary plant facilities including existing condition, operation characteristics, near term repairs and maintenance requirements, estimates of long term project repairs, and a schedule of activities.

The cost estimates and schedules prepared for St. Anthony Falls will serve as a basis for additional financial evaluations of development options for the Hennepin Island Hydro plant. This planning is critical for monitoring the site activities and planning future development.

SITE DESCRIPTION

The St. Anthony Falls facility located in Minneapolis, Minnesota, was built in the 1870's, with the most recent improvements completed in 1954-1955. The project is significant to local history and is a focal point of recent riverfront development. Figure 1 is a plan of the site that illustrates each of the primary project features. The facility contains many components that were part of the original 1870 era structure.

The site's historic significance was a prime factor when considering alternatives. Essentially all project alternatives required that existing alignment of the major structures be maintained, and in most instances the existing structures were required to be protected. Currently, there is no powerhouse at the lower dam, and it is not producing power. The site is being evaluated for redevelopment. Redevelopment of hydro power at the lower site could be accomplished in conjunction with work at the upper site. All of these factors were considered in a separative study.

The St. Anthony Falls project consists of two separate dams, 1) an upper dam that has a head of approximately 49 feet and 2) the lower site, which has a head of approximately 25 feet. The upper dam includes spillways, non overflow sections, powerhouses, island, and a U.S. Government Lock and Dam. The lower dam includes earth dams, and a U. S. Government Lock and Dam. When it was built, the project provided hydro power for producing electricity and driving mills for grinding wheat for flour and saw mills that manufactured wood products.

Upper Site

The primary features of the St. Anthony Falls upper dam include: 1) two hydro plants, (Main street (retired) and Hennepin Island Hydro), 2) Hennepin Island, 3) 270 feet of non-overflow sections, 4) 2725 feet of overflow spillway, 5) St. Anthony Falls Hydraulic Laboratory (University of Minnesota), and 6) two abandoned wasteways. The overflow spillway is comprised of nine sections, each constructed at different times using various cross sections.

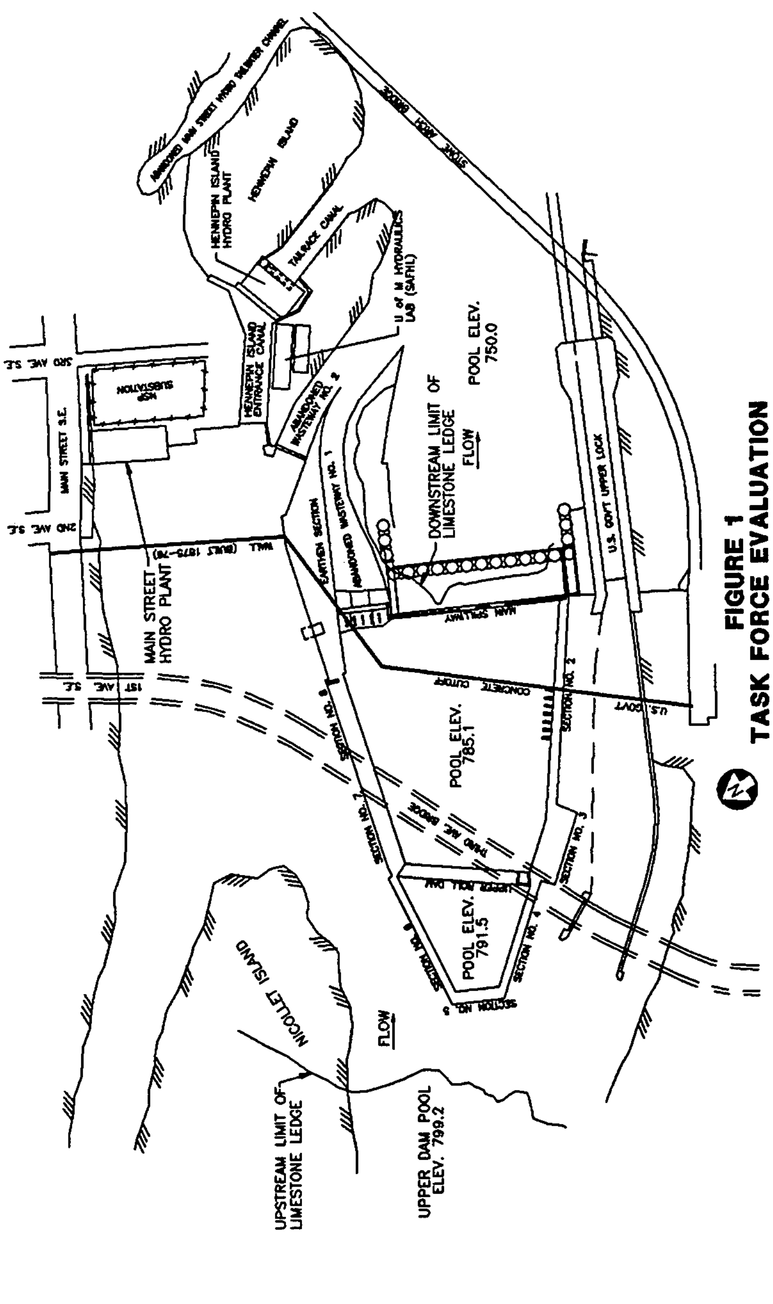


FIGURE 1
TASK FORCE EVALUATION
St. Anthony Falls

The St. Anthony Falls upper dam was constructed at the downstream edge of the upstream advancing ledge of limestone. Historically the underlying sandstone was being eroded by the river flows with the limestone layer being undermined and breaking away. The downstream edge of the limestone ledge was advancing upstream at a maximum rate of approximately 20 feet per year. The limestone formation is approximately 1,200 feet long and would have been completely lost in approximately 60 years. Construction of the dam was critical to protect the underlying sandstone from further erosion. If the sandstone was allowed to continue eroding, the falls would have advanced upstream quite possibly without significant restraint. Protection of the downstream edge of the limestone ledge and underlying sandstone was required for all options.

PROJECT DESCRIPTION

The work for completing the plant assessment consisted of: reviewing existing documentation (reports and drawings), performance of site inspections, evaluation of component conditions, estimation of maintenance and repair requirements, projection of long term maintenance requirements, (including possible replacements), and preparation of cost estimates and tentative schedule for all of activities. The evaluations were performed to consider the options for maintaining a financially viable facility. Factors such as staffing requirements by NSP for operations, timing of critical and noncritical repairs, improvements or upgrades, and coordination of activities were addressed.

Options for redevelopment of the St. Anthony Falls sites are being considered by (NSP). The assessment presented information regarding the upper development and the existing Hennepin Island Plant to support the overall evaluation of the various options for the facility, and serves as a planning document for future activities. The methodology used for the report included:

1. Collection and review of the existing civil, electrical, and hydro-turbine information for the facility.

2. Performance of basic site inspection of the dam/powerhouse components, turbine equipment, generator, and other electrical appurtenant equipment. The purpose of the inspections was to update, verify, or supplement the available information.

3. Development of a schedule and cost estimates for the repair, maintenance, or upgrade of the facility. The costs were separated into the pre-license period, and two periods (10 and 20 year), during the remainder of the license.

The upper dam includes the Hennepin Island Hydro (HIH), which was separated from the other components because several options for the plant were being considered. These options included maintaining the existing capacity, upgrading the capacity, and retiring the facility. The requirements of many of the remaining components were unaffected by the option selected for HIH.

SITE INSPECTIONS

Site inspections were performed by Barr Engineering Co., L & S Electric, Inc. and American Hydro Corp. to determine the condition of the existing electrical and hydro-turbine equipment. The following paragraphs describe the activities performed by each consultant.

Electrical

Electrical site work included the performance of power gate tests, stator megger tests, polarization tests, rotor megger tests, and rotor drop tests. These tests, in conjunction with a site inspection and interview of NSP maintenance personnel, provided a basic understanding of the condition and the efficiencies of the equipment through the normal operating range. Overall the electrical equipment was found to be in good condition, although some components needed repairs. The inspection indicated that generator faults can be expected in the near future. To restore confidence in the existing control and protection equipment, a retesting and recommissioning program was recommended. Completion of these work tasks should provide a reliable electrical system that requires only routine maintenance during the years of the current license as well as the life of a new license. The existing generators were found to have sufficient capacity to provide service for upgraded hydro-turbines if the generators were rewound.

Hydro-Turbine

Hydro-turbine site work included performance of a visual inspection of Unit 1 (which was dewatered for maintenance), inspection of the intake and tailrace areas, and evaluation of the power gate tests performed by L & S Electric. The inspection showed nothing unusual. The intake includes an intake canal that is approximately 330 feet long from the headgates to the powerhouse. The intake walls are formed of limestone blocks placed on bedrock. Units 1-4 are horizontal double runner francis turbines that are all similar in condition and have similar operating characteristics. Periodic maintenance has been required on the turbines for Units 1-4 and the condition of Unit 1 is consistent with previous reports. Unit 5 is a vertical kaplan unit that requires operation under much different characteristics than Unit 1-4. Nothing significant was noted about Unit 5 except that given its setting the gate opening is limited to prevent operating under cavitating conditions. The inspection revealed that all units are operated near optimum. A statistical analysis of the setting and performance of Units 1-4 indicated that the maximum output could be increased significantly, primarily by the installation of modern design runners. Unit No. 5, however, given its age and maintenance history likely cavitates more than normal. Statistical analysis indicated that with a modern runner design the output would have minimal increases. However, maintenance requirements would be reduced.

The potential increase in power production was compared to the maximum increase from the rewind and refurbishing of the existing generators and related equipment.

Civil Components

Civil site work included performance of a limited inspection of the primary civil facilities to supplement the results completed for previous FERC Part 12 inspections. The information contained in the previous reports provided the primary basis for determining the condition and needs of the various project facilities. The condition of the facilities varied considerably from near new components requiring minor repairs to facilities that are near the end of their service life and require replacement. The condition of each component was evaluated separately and the maintenance requirements for replacement through the life of the license were assessed.

The site inspection and evaluations provided a review of the condition and serviceability of each of the project components. The component evaluation allowed the development of cost estimates to implement repairs and improvements. Cost estimates to correct existing known deficiencies were prepared and can be used with a high degree of reliability. The scope and nature of future activities were determined for each major project feature. The recommended required repairs were based on professional judgement and experience. Where major rehabilitation

is being performed, the amount of repairs required is significantly reduced. However, where a component is being repaired to a lesser extent, the frequency of repairs as well as the scope can be increased significantly. The component descriptions, estimated costs, and tentative schedules were tabulated as noted below.

COST ESTIMATE OF REPAIRS AND REPLACEMENTS
ST. ANTHONY FALLS UPPER DEVELOPMENT

Components	Total Quantity	Unit	Tentative Repair Schedule (Years)			Component Section Reference	Comments
			-8± to 0	0 to 10	11 to 30		
Tailrace Canal Concrete Replacement Wall (right) Flare Left Side (riprap) Flare Left Side (excavation)							
Wasteway No. 2 Rock Filled Timber Crib (modify)							
Earthen Section Boat Ramp Access Concrete (repair) Boat Ramp Access Timber Stoplogs Repair Wood Buildings Seepage Cutoff Artificial Rock Facing (pile cap) Toe Drain (2 ft x 2 ft)							

The evaluations indicated that the existing electrical equipment could be upgraded sufficiently through refurbishing to match the potential upgrade from the turbines. This clearly indicated that the option of upgrading the plant was worth further consideration. The comprehensive evaluation allowed for determining these facts as well as exploring the impact an upgrade would have on the civil components of the facility.

The St. Anthony Falls upper dam site is a complex site containing a diverse group of components that each uniquely impacts the options and direction of the project. Careful planning for implementation of repairs is required. Coordinating electrical, civil, and hydro-turbine activities to minimize project down time and duplication of efforts was a priority.

SUMMARY

The plant assessment revealed that due to the varying ages, conditions and functions of the project components, the urgency, scope, and timing of the repairs vary considerably. Some components have provided years of service and have essentially exhausted their service life. Other components are in good condition, requiring only preventative maintenance. These varying conditions fit well into the format of the assessment, which allows for coordination of activities for economy of activities.

The specific conclusions developed for the St. Anthony Falls project are not particularly significant to those not involved on this project. What is important is the basic premise underlying the project and the process. Formulation of a technically competent project team to

perform a comprehensive evaluation of all components of the facility was critical to a successful completion of the work.

The plant assessment report presents the estimated costs to correct known deficiencies, possible future repairs, upgrades, and modifications. The cost estimates and tentative schedule of repairs provide a basis from which NSP can develop future plans.

The report also provides information essential to the completion of relicensing activities and evaluation of alternatives for facility upgrades.

Plant assessments have been performed on ten of the NSP facilities by the this group of consultants. The reports have been used to formulate plans for project rehabilitation. With updating and adjustments, the cost estimates and schedules are still being used.

REFERENCES

"Application for a New License Major Project - Existing Dam for the St. Anthony Falls Hydro Redevelopment Project (FERC Project No. 2056), submitted by Northern States Power Company, November 1986

Barr Engineering Co., "An Examination pursuant to Order No. 122 as Set Forth in Subpart D of Part 12 of the Federal Energy Regulatory Commission for the St. Anthony Falls Hydro Project FERC No. 2056, September 1988

Barr Engineering Co., "Technical Appendices" to the 1988 FERC report, September 1988

American Hydro Corporation, "Hydroturbine Life Extension and Upgrade Analysis for Hennepin Island Plant", March 16, 1992

L & S Electric, "Plant Evaluation for St. Anthony Falls - Hennepin Island Hydroelectric Plant", March 1992

Barr Engineering Co., "Plant Assessment St. Anthony Falls - Upper Development Hennepin Island Hydro Plant FERC License No. 2056", 1992

Puget Power's Hydro Modernization Project,
a Diversified Approach

Michael J. Haynes, P.E.[1]
David F. Webber, P.E.[2]
John B. Yale, P.E.[3]

ABSTRACT

Puget Sound Power and Light's (Puget Power) hydroelectric generation system can be summed up in one word, diversity. A wide range of project layouts includes the world's first totally underground powerhouse, impulse and reaction turbines, stationary field generators and a myriad of control technology. In 1991, a concept was formulated to systematically modernize each project from the ground up. The challenge faced was to establish a comprehensive program designed to prioritize 6 power plants, specify a quasi-standard control methodology and implement installation and start up; all with minimal impact to system energy production. This paper presents a comprehensive discussion of the conceptual design process, specification development and project planning.

INTRODUCTION

Puget Power is an investor-owned electric utility which has been operating and maintaining hydro electric projects in Washington State since 1898. The hydro

---

[1]Senior Engineer, Engineering Services Department, Puget Sound Power & Light Company, P.O. Box 97034, OBC-11S, Bellevue, Washington 98009-9734

[2]Engineer, Engineering Services Department, Puget Sound Power & Light Company, P.O. Box 97034, OBC-11S, Bellevue, Washington 98009-9734

[3]Senior Control Engineer, Stone and Webster Engineering Corporation, 7677 East Berry Avenue, Englewood, Colorado 80111-2137

system consists of five (5) projects located throughout the western portion of the State of Washington. This includes three (3) run of the river projects and two (2) reservoir projects. The projects range in age from 33 to 93 years and have a combined output of approximately 335 MW. Currently, two projects, White River and Nooksack Falls, are undergoing the federal license process and another, Snoqualmie Falls, is a federally licensed project undergoing the relicensing process.

The concept of modernization is an evolving philosophy which is being developed to provide a proactive approach to some specific hydro issues (and constraints) faced by Puget. Primarily these include: 1) Turbine and generator upgrades, 2) The potential impacts of future (license driven) operational and environmental requirements, and 3) Benefits associated with improved efficiency, unit dispatch and enhanced control and monitoring.

Early in the planning process, the decision was made to approach the project from a long-range viewpoint. The long range perspective was implemented as a means of maintaining continuity from project to project. This allowed Puget to consider the upgrade of its hydro system as a whole, thereby promoting consistency, while allowing flexibility of the design on a plant by plant basis. In addition, other benefits of long range planning could be realized. An example of this is the experience factor, which will improve start-up efficiency from project to project. Completion of the first project would also allow Puget's engineers and operators to gain valuable information and construction experience in order to prepare for subsequent upgrades which will likely involve more demanding construction schedules.

BACKGROUND

Upon initial review of its hydro system, Puget developed a set of criteria for the Hydro Modernization Project. This development determined that such a project must not only meet the challenge of external constraints, but also meet the challenge of time.

1. Modernization would proceed on a plant by plant, unit by unit basis, with an emphasis on standardization where possible.

2. The upgraded projects must possess the flexibility to interface with Puget's existing Energy Management System (EMS) and any future upgrades of the system; and

3. All modernization work would be reviewed on a case by case basis to determine its cost-effectiveness.

HYDRO SYSTEM OVERVIEW

Baker River Hydroelectric Project

The Baker River Project (FERC Project #2150) consists of two (2) reservoir developments (Lower Baker and Upper Baker) under a single Federal Energy Regulatory Commission (FERC) license which expires April 30, 2006.

Lower Baker River Project

The project houses a single Francis style turbine which is rated 80,000 hp, 236 feet, 163.6 rpm. Maximum discharge at rated output is 4,034 cfs. The generator is rated 73,600 kVA, 1.0 pf, 80°C. The unit was rewound to its present rating in 1983.

Current Project Operation

The unit is controlled remotely from Puget's Eastside System Operations Center (ESO) in Redmond, WA. It can be started and synchronized automatically and loaded using raise/lower controls for MW and VARs.

Inflow to the Lower Baker dam comes from the Upper Baker plant discharge and some minor side flows. The unit is normally operated in a forebay control mode, but can also be operated in a synchronous condenser mode. The latter mode is rarely used.

Proposed Project Modifications

Work is currently in progress to study the feasibility of adding an additional unit and to evaluate potential efficiency/capacity improvements of the existing unit.

Upper Baker River Project

The plant features two vertical shaft Francis style turbines each rated 70,000 hp, 285 feet head, 200 rpm. Maximum discharge of each unit at rated output is 2,542 cfs. The turbines power two vertical generators each rated 52,400 kVA, 0.9 pf, 13.8 kV. These units were rewound to their present ratings in 1990 and 1989, respectively.

Current Project Operation

The plant is controlled remotely from Puget's ESO. The units can be started and synchronized automatically. The project operates in a forebay control mode for a major portion of the year and in flow control during the flood season.

Proposed Project Modifications

A turbine upgrade feasibility study is ongoing. This study is designed to explore possible improvements in efficiency and capacity. Modifications to the existing station service equipment are also planned.

Electron Hydroelectric Project

The powerhouse contains 4 double overhung impulse turbines. Units 1-3 are rated 7,500 hp each, 865 feet head, 225 rpm. Unit 4 is rated 10,000 hp, 865 feet head, 257 rpm. Maximum discharge for the plant at rated output is 380 cfs (Units 1-3 = 90 cfs each, Unit 4 = 110 cfs).

The generators for Units 1-3 are rated 6,000 kVA each, 1.0 pf, 2300 V. Unit 4 is rated 9,375 kVA, 0.8 pf, 2300 V. Solid state excitation for all four units was added in 1981.

Current Project Operation

The project operates as a run of the river plant, adjusting load to maintain forebay level control based on the level of water in a 10 mile long flume. The Electron Project is currently operated remotely from the White River Generating Station. Remote control is limited to raising and lowering of unit load, reactive output and diversion flow.

Load adjustments on this project are quite restricted due to the volume of water in the flume at any one time and the lack of sufficient storage area to accommodate it. Because of this, the plant is very sensitive to load rejections. All four units have the capability to be run as a synchronous condenser.

Proposed Project Modifications

Ongoing studies include review of turbine designs and unit capacities in order to assess potential upgrade options. Preliminary evaluations indicate efficiency improvements of up to 15% can be achieved through turbine modifications, including runner replacement.

Nooksack Falls Hydroelectric Project

The Nooksack Falls Hydroelectric Project is located on the North Fork of the Nooksack River, approximately one-half mile below Nooksack Falls in Whatcom County, Washington. In February, 1982, Puget Power submitted a license application to the FERC for the Nooksack Project. The issuance of this license is pending.

The plant features one impulse turbine rated 2,547 hp, 195 feet head, 200 rpm. Maximum discharge at rated output is approximately 220 cfs. The generator is rated 1,700 kVA, 1.0 pf, 2200 V. The unit is separately excited, using a water-powered, belt driven DC generator.

Current Project Operation

The Project operates in a run of the river mode. The plant operates without remote control capability and is not presently interfaced with Puget's Energy Management System (EMS).

Proposed Project Modifications

Modifications to the existing facilities will be dependent upon the issuance of the FERC license. The proposed project would feature a new powerhouse, dam and flowline.

Snoqualmie Falls Hydroelectric Project

The Snoqualmie Falls Project (FERC Project #2493) consists of two (2) powerhouses (Plant 1 and Plant 2) under a single FERC license. Puget Power has recently submitted an application for the relicense of the Project with the FERC. The existing project license will expire on 12/31/93.

Plant 1 features four horizontal impulse turbines rated 2,200 hp each, 251 feet head, 300 rpm. Maximum discharge on the individual units is approximately 115 cfs each. Unit 5 is a horizontal Francis style turbine rated 10,000 hp, 260 feet head, 300 rpm. Maximum discharge for this unit is approximately 345 cfs.

Plant 2 houses two units. Unit 1 is a horizontal Francis style turbine rated 10,000 hp, 270 feet head, 360 rpm. Unit 2 is a newly replaced, vertical Francis style turbine capable of producing 34,500 hp, 270 feet head, 360 rpm. Maximum discharges are 470 cfs for Unit 1 and 1,225 cfs for Unit 2.

Units 1-4 in Plant 1 are horizontal shaft, stationary field generators which operate at 2000 V. The unit ratings are as follows: Unit 1 - 1,500 kVA, Unit 2 - 1,800 kVA, Unit 3 - 1,500 kVA, and Unit 4 - 1,500 kVA, all at 1.0 pf. Unit 5 is rated 6,220 kVA, 0.9 pf, 2000 V. Units 1-4 are excited on a common bus using a waterwheel-driven DC generator. Unit 5 is excited using a motor-driven DC generator.

Unit 1 in Plant 2 is 12,300 kVA, 0.80 pf, 6900 V and was rewound in 1956 to its present rating. Unit 2 is rated 22,500 kVA, 0.9 pf, 6900 V.

Current Project Operation

Snoqualmie Falls operates as a run of the river plant. However, the Project is required to pass a certain volume of water over the top of the dam at all times and more closely resembles forebay control with no appreciable storage. Once the units are placed on line locally, control is turned over to the load dispatchers who operate the Project remotely from Puget's ESO. A central control house exists on the Project to allow for simultaneous manual control of both plants from one location when the need arises.

Proposed Project Modifications

Based on the modifications proposed in the license application, the Project will feature: a new, single unit (~ 9000 kVA) in Plant 1 and 3 units in Plant 2. This would include rehabilitated Units 1 & 2, and a new Unit 3. Modifications are also proposed for the dam structure and the entire Plant 2 flowline.

White River Hydroelectric Project

In November, 1983, Puget Power submitted a license application to the FERC for the White River Project (#2494). This application addressed the existing facilities and upgrades/additions to portions of the Project which would result in improved use of the water resource. The issuance of this license is pending.

The Project consists of a diversion dam, flowline, reservoir, forebay structure and powerhouse.

The powerhouse contains four units consisting of two pairs of identical turbines. Units 1 and 2 are horizontal Francis style turbines rated 18,000 hp each, 440 feet head, 360 rpm. Units 3 and 4 are horizontal Francis style turbines rated 23,000 hp each, 440 feet, 360 rpm. Maximum plant discharge is approximately 1,970 cfs.

Generators for units 1 and 2 are rated 16,300 kVA each , 0.92 pf, 6600 V. Units 3 and 4 are rated 25,000 kVA each, 0.8 pf, 6600 V. Solid state exciters for all four units were installed in the early 1980's.

Current Project Operation

The White River Project operates as a combination run of the river/reservoir plant. Due to recreational requirements, the project operates in a run of the river mode during the summer months to hold the reservoir level constant. The plant is used for peaking during the winter months.

White River is a local, manual plant without remote capability. Operators maintain control locally 24 hours per day. Due to the proximity of the projects, the Electron Project is operated remotely from White River through the EMS.

Proposed Project Modifications

As part of the pending FERC license application, modifications to the dam, intake structure and fish handling facilities are proposed. Also, in conjunction with the newly constructed pipeline, a new in-line powerhouse with a capacity of approximately 15,000 kVA is proposed.

Modifications to the existing powerhouse will include: turbine and governor upgrades, generator replacements, station service upgrades and possibly tunnel/penstock upgrades. Preliminary investigations of these possible modifications have been completed. Detailed feasibility assessments of various upgrade alternatives are ongoing.

HYDRO SYSTEM UPGRADE PHILOSOPHY

In addressing unit/system upgrades, Puget Power has developed a basic plan which is designed to identify modernization elements specific to each location. This approach involves:

1. Evaluation of the condition of each project and each unit within that project.

2. Performing tests and studies as necessary to evaluate unit upgrades and potential additions for improved resource utilization, increased efficiency and capacity, and increased reliability and flexibility.

3. Performing studies and developing specifications for a unified control system.

4. Planning and scheduling upgrade and modernization work in a way to limit outages and unit down time.

Existing projects are evaluated both physically and economically in order to justify improvements. The level of feasibility ultimately defines a particular modernization scope for each individual plant, with all plants being unique.

The path to any study of upgrades, for the purpose of resource utilization and allocation, requires a methodology which is consistent within a given region. Parameters established by State and local governments and resource agencies dictate balancing guidelines which must become the base line of any study.

In order to address this, the concept of balancing has developed into the foundation for the hydro licensing process. In this venue, balancing looks at the relative importance of power and non-power project aspects.

Non-power project aspects include recreation, fisheries, aesthetics, wetlands, water quality, water supply, historical/cultural, flood control and air quality, to name a few.

Instream flow requirements, the heart of most non-power issues, may ultimately define the base line for power related project considerations. Furthermore, a control system specification must address any such operating constraints in order to accurately define closed loop control algorithms.

Another key aspect of an existing project is the vintage of installed equipment. Generally speaking, hydraulic turbines manufactured prior to 1960 will exhibit reduced capacities and efficiencies when compared to modern designs for the same conditions. Efficiency improvements can be especially attractive since more generation can be obtained with the same amount of water. The difficulty with this type of assessment is establishing economic feasibility. Additional generation due to efficiency improvements alone may not prove economically viable. Utilities, therefore, must consider less tangible aspects such as reliability, operation costs and maintenance costs in order to substantiate rehabilitation.

The role of modernization in establishing feasible water resource options represents an evolving technology aimed at addressing environmental and economic accountability. It is the responsibility of the hydro system owner or developer to satisfy sound engineering principles, while responding to imposed constraints related to water resource development.

Modernization, therefore, is a tool which allows the best technical optimization of a resource in conjunction with allowing designers to factor in environmental parameters. Issues such as instream flow, ramping rates, reservoir operating curves, flood control and aesthetics can be effectively managed through proper modernization planning.

CONTROL SYSTEM

Several fundamental parameters must be considered when developing a control system for such a diversified system of hydro plants. (PSPL; June, 1992) These include:

1. Interface with the existing Energy Management System (EMS) including planned upgrades expected to be completed prior to complete implementation of the hydro control system.

2. Upgradability, product life, and vendor reliability.

3. Standardization from project to project.

One of the major restrictions on the control system with respect to the modernization program is Puget's implementation schedule, which may span ten years. It is desirable to install automated control systems as each unit and station is rebuilt or each new project is constructed. Therefore selection of a vendor and an architecture that has demonstrated an ability to provide consistent services and systems over time, and show the desire and ability to provide consistent services in the future, becomes extremely important.

A significant benefit of installing an automated control system is enhancing efficient utilization of the individual projects. Each project will be provided with control functions to operate at best efficiency based on a variety of parameters. Since the hydraulic conditions and parameters of each project are different, different operating modes will be utilized for each project. (PSPL; August, 1992) These would include:

1. Level Control
2. Flow Control
3. Peaking based on generation schedule
4. Peaking based on reservoir level constraints

CURRENT STATUS

To date, and in addition to development of the Hydro Control System Conceptual Design and Technical Specification, the tasks that have been completed include:

White River Turbine Upgrade Study

This study addressed the potential for installing replacement turbine runners in the existing units in order to increase efficiency and capacity. The results of this study showed a potential for increased capacity of between 8 and 35 % over the existing units, with some efficiency improvements. The increased capacities will be useful in better matching the turbines with the currently oversized generators. The generators would be rewound and the stator cores restacked in conjunction with any turbine upgrade. (AHC 1992)

Electron Turbine Upgrade Study

An overview of this project identified integrally cast replacement runners, which would improve efficiencies by up to 15 %. Additional improvements would

include adjustments to the turbine settings and rehabilitation of the generators.

Lower Baker Turbine Optimization/Upgrade Study

A potential unit capacity increase of up to 10 % has been identified for the existing unit, and an evaluation of additional unit capacity is currently under way. This project is also being evaluated on the basis of hydraulic optimization of available water. The intent of these studies is to develop an economic plan for improving the overall project operating efficiency.

Snoqualmie Falls Turbine Upgrade Study and Runner Replacement

A recent turbine upgrade at the Snoqualmie Falls Project attained an increase in turbine horsepower from 28,000 to 34,500. A study of upgrade potential was completed prior to development of a feasibility assessment. (AHC 1991)

CURRENT STATUS

The Hydro Modernization Project will continue into 1993 with additional studies planned to determine the hydraulic sensitivity for the White River, Electron and Baker River Projects, completion of final design for the manual controls and protective relay systems for the existing White River powerhouse, and the completion of an technical specification for the supply and installation of the future hydro control system.

REFERENCES

American Hydro Corporation, "Engineering Study of Upgrade Potential, Snoqualmie Falls Project", January, 1991.

American Hydro Corporation, "Engineering Study of Upgrade Potential, White River Project", February, 1992.

Puget Sound Power and Light Company, Stone and Webster Engineering Corporation, "Conceptual Design Report - Control System for Hydro Modernization, June, 1992.

Puget Sound Power and Light Company, Stone and Webster Engineering Corporation, "Specification for Automated Control System", August, 1992.

The Monroe Street Hydroelectric Project Redevelopment
A Legacy of Power 1889 - 1992

Michael J. Finn[1/]
Steven G. Silkworth[2/]

Abstract

The Monroe Street Hydroelectric Project is being redeveloped after 100 years of successful operation. The redeveloped plant will double the power output of the original plant from 7.0 to 14.0 MW using the same head and existing water rights. The project's dam, intake, and upper penstock were replaced for the 1974 World's Fair. Redevelopment of the lower penstock and powerhouse began in 1989 and was completed in July 1992. The plant was commissioned on December 1, 1992.

This paper discusses some of the more interesting aspects of the project—a three-way construction contract split; a two-stage cofferdam; environmental work; a low submergence intake; a virtually unnoticeable powerhouse; and future provisions for full development of an additional 50 MW at the site.

Introduction

The Monroe Street Hydroelectric Project was developed by the Washington Water Power Company to celebrate a triple centennial. It was the simultaneous 100-year anniversary of: the original hydroelectric plant, the Washington Water Power Company, and the State of Washington, all which occurred in 1989.

This project is located in the heart of the City of Spokane, Washington, at the base of the Spokane waterfall and adjacent to Riverfront Park, site of the 1974 World's Fair. The powerhouse is nestled under the landmark Monroe Street bridge from which the project derives its name and is overlooked by the City Hall building, just 400 feet to the east. See the location map Figures 1 and 2.

1/ Project Manager, Ebasco Services Incorporated, Seattle, Washington.

2/ Project Manager, Washington Water Power Company, Spokane, Washington.

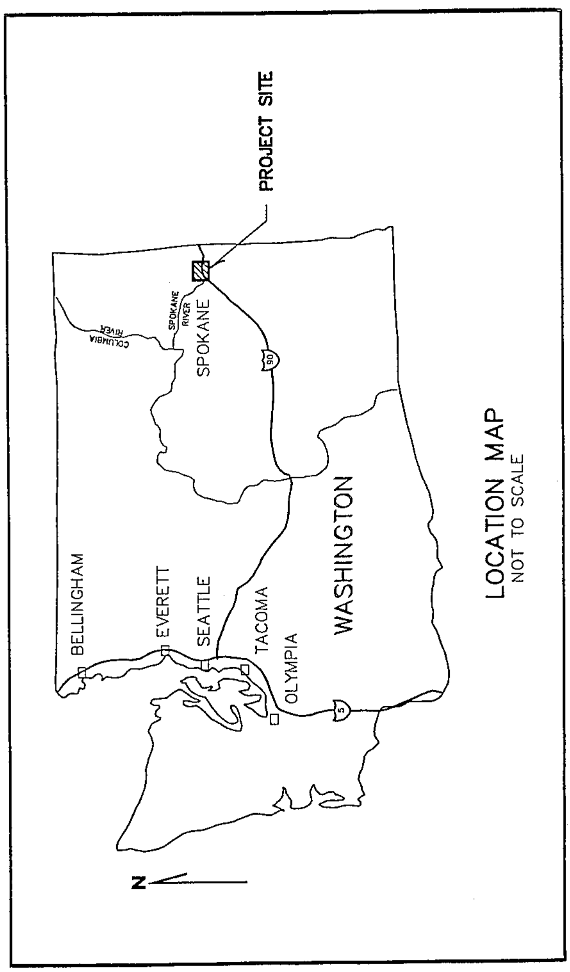

PROJECT SITE
SPOKANE RIVER
COLUMBIA RIVER
SPOKANE
90
BELLINGHAM
EVERETT
SEATTLE
TACOMA
OLYMPIA
5
WASHINGTON
LOCATION MAP
NOT TO SCALE
N

Figure 1.

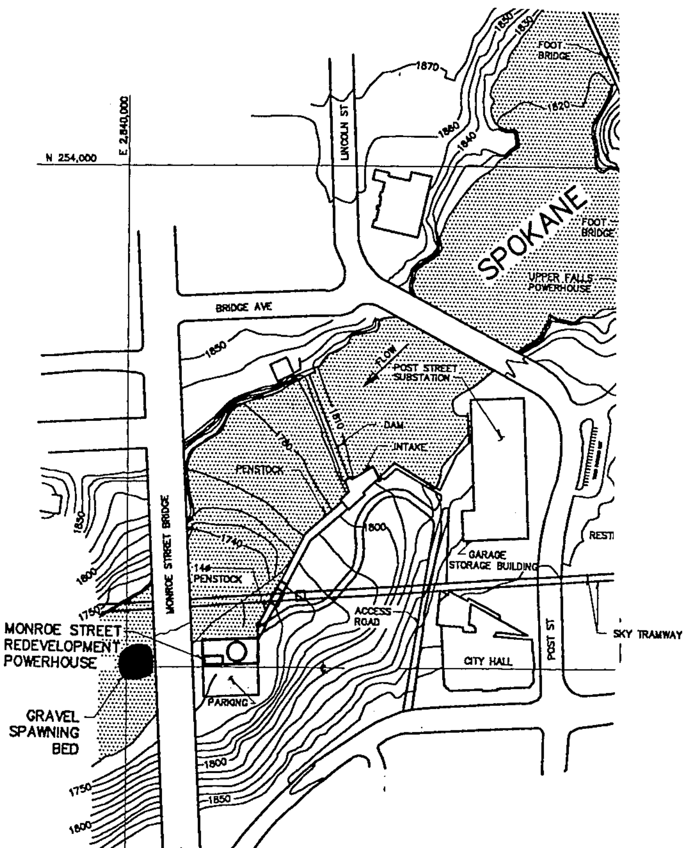


Figure 2 Local Map

The project has several unique features which were interesting in development. The following is a summary of further discussion in this paper.

1. Plant output was doubled from 7 to 14 MW using the same head and existing water rights.

2. The original plant's five small units were replaced by one 14 MW vertical Kaplan unit.

3. The powerhouse is totally below grade allowing the project area to be beautifully landscaped into what is now the site of Huntington Park.

4. The powerhouse is situated to allow future development of a 50 MW plant immediately adjacent to the current one.

5. The FERC License amendment was fast-tracked and accepted within 3 months of submittal.

6. Three separate contractors were used to expedite construction:

   a. The first for cofferdam and demolition;
   b. The second for powerhouse excavation; and
   c. The third for general construction.

6. A two-stage cofferdam was used to facilitate construction in the very deep narrow river.

7. The construction site was very small and tight, necessitating an offsite storage yard, erection area, and office. Access roads were moved to match the focus of construction.

8. A wedge-shaped side entrance intake.

The following is a table of:

SIGNIFICANT PROJECT DATA

Installed Capacity	14 MW
Rated Head Net	68 feet
Design Flow	2,600 cfs
Maximum Flow	3,000 cfs
Turbine Type (full, vertical)	Kaplan
Synchronous Speed	200 rpm
Runaway Speed	(510) rpm
No Blades	5
No Wicket Gates	24
No Stay Vanes	24
Runner Diameter	10.5 feet
Spiral Case Inlet Diameter	14.0 feet
Generator Rating	15.7 kVA
WR^2	100,000 lb/ft^2
Voltage	13.8 kV
Amps	606.6
Power F	0.95
Penstock - Diameter	14′-0
- Length	420 feet

Full Site Development

The first phase of the project began with a preliminary design assessment of the full potential of the site. Since average flow in the river is 6,800 cfs, the site was deemed to support 64 MW in two plants. The first, called the Redevelopment, was constructed. The second, a 50-MW plant in one full Kaplan unit, was not developed at this time, as the client had other more economical sources of power to develop first. However, provisions and space for the second powerhouse's construction at a future date were incorporated into the design of the Redevelopment project.

Three-Part Contract Split

For the 14-MW Redevelopment project, three construction contracts were used to expedite project schedule and meet required windows in the river's regulated flow pattern.

To begin first-stage cofferdam construction in the low-flow month of August, a first contract was let for this item and also included demolition of structural elements, construction of access roads, and the sediment pond. As soon as cofferdam and demolition work were completed, a second contract was let for general rock excavation of the powerhouse and lower penstock.

While these two contracts were being performed, a general construction contract for the remainder and majority of the work was being bid, evaluated, and awarded.

This split of the work into three parts including two small, initial packages left adequate time for design and bidding of the general construction contract, which was the bulk of the project, and thereby avoided a delay of over one year in the start of construction.

Environmental

Erosion and sediment control was provided by routing all site runoff in the construction area through a three-stage settling pond, 25 feet wide and 75 feet long. Special provisions were made to avoid turbulence at the inlet; flocculents were used to accelerate settlement; frequent cleaning was required; and the outlet pipe was well filtered.

Large gravel-filled bags were used to form the first stage of cofferdam construction. The gravel used was a blended 1/2- to 1-1/2-inch size specifically suited for spawning of local fish. As the cofferdam was removed, the gravel was flushed into a nearby area just downstream of the project to provide a spawning bed.

Cofferdam

The cofferdam was a major challenge. The powerhouse and tailchannel are located in a narrow constricted part of the river just adjacent to a 70-foot-deep hole at the base of a waterfall. A cellular sheet pile cofferdam just wouldn't fit at this site. The river was too narrow and too deep. In addition, sheet piles would have been difficult to seal against the blocky irregular riverbed. The river width is also constrained at the powerhouse section by two massive piers for the Monroe Street bridge, 100 feet overhead.

To accommodate the tight constraints at this site, a very narrow concrete gravity dam section was used in two stages. The first stage was built virtually in the dry and enclosed only the powerhouse. It was in place for almost 18 months and was designed needed to withstand the full seasonal range of water levels. The second stage was similar in section and enclosed only the tailchannel. It was in place for only the 2 months required to construct the cofferdam and excavate the tailchannel. The concrete sections were ideal to seal against the blocky irregular basalt river bottom.

Intake Structure

The unique intake structure uses a side slot wedge-shaped inlet to cut normal submergence requirements in half. It was desirable to have the intake no lower than the base of the dam. To do so left only 5 feet of submergence available where a conventional intake requires nearly 20 feet of submergence. An elevation and section of the wedge-shaped, side-slot intake is shown on Figures 3 and 4. The hydraulics of this intake are based on a paper by Milton Sulzman, titled Water Intake Designed for Wide Distribution of Flow, Engineering News Record, December 16, 1926. The intake is arranged in such a way that the velocity through each of the side slots is equal and significantly lower than in the pipe just downstream of the intake. This long low alligator-type intake produces more than double the head loss (through turbulence) than is produced by a conventional intake; however, it is useful where adequate submergence is just not available.

Powerhouse

It was desired to have the powerhouse virtually inconspicuous, as it was at the focal point of a new city park approximately 400 feet from City Hall. To do this, the powerhouse was set deep into rock totally below grade with its top deck at final grade. The roof deck received a 4-inch architectural topping slab of exposed aggregate around a sunburst pattern formed of double rows of brick. For watertightness, reinforcing bars were spaced at 6 inches oc in the downstream walls and roof deck.

The powerhouse was a conventional reinforced concrete structure housing a single 14-MW vertical Kaplan unit. A plan and transverse section of the powerhouse are shown on Figures 5 and 6. There is no permanent crane provided. Major equipment for this project is of a size that it can be handled by a rented, 175-ton, mobile crane.

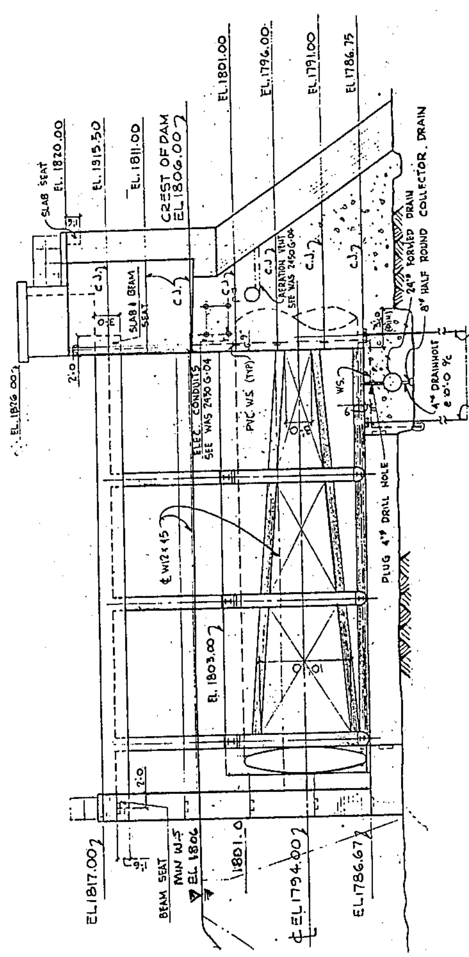


Figure 3. Intake - Elevation

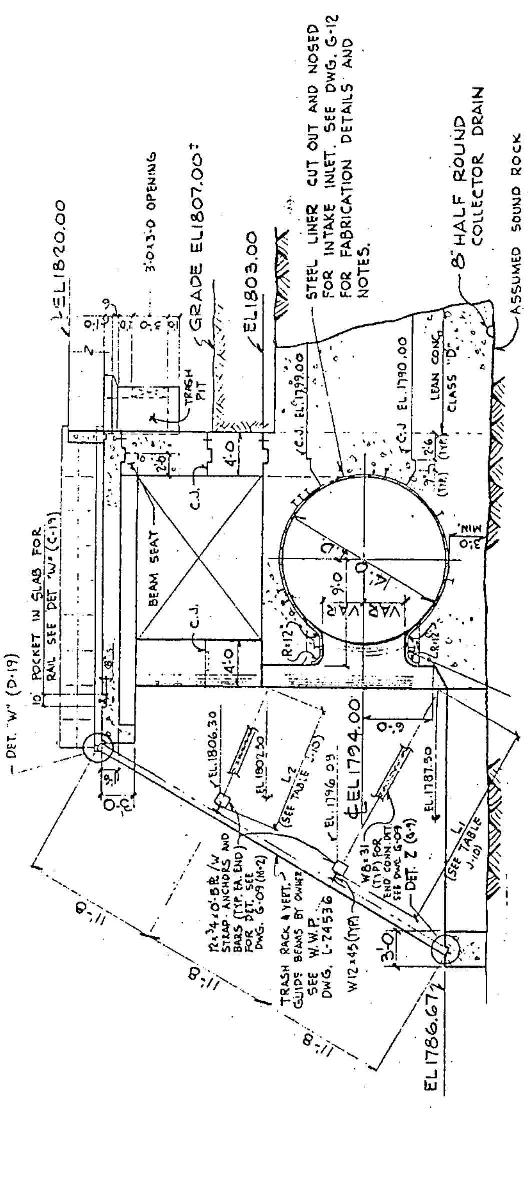


Figure 4 Intake Section

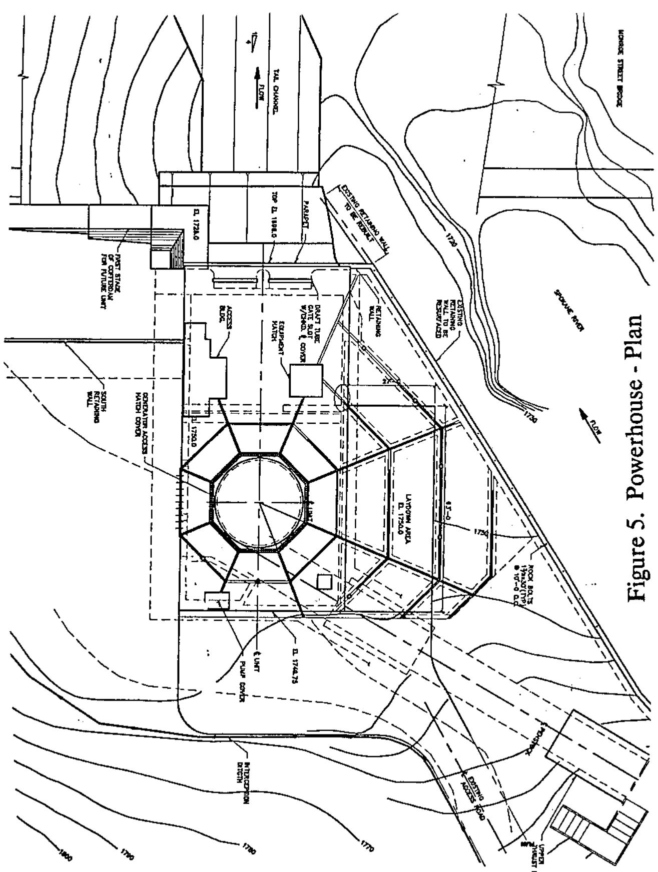


Figure 5. Powerhouse - Plan

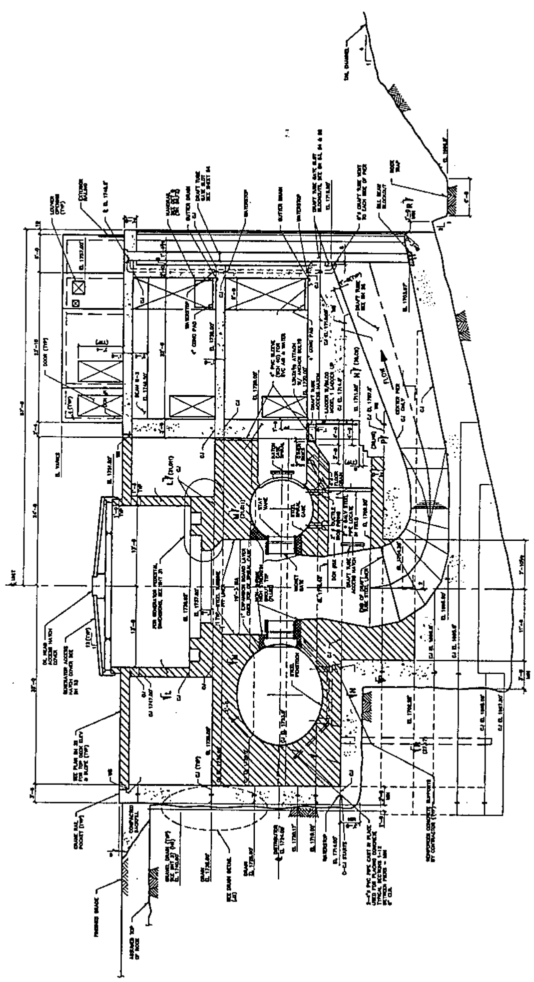


Figure 6

Powerhouse Transverse Section

The following is significant data about the:

Sizes and Weights of Major Equipment

Item	Diameter	Weight	Notes
1. Runner	10.5 ft	33	5 blades
2. Shaft	19"	--	L = 25'-0
3. Generator Rotor	12'-5"	115 K	With shaft
4. Generator Stator	20'-0	75 K Total	Shipped in two pieces

Project Development Team

Developer	Washington Water Power Company
Engineer & CM	Ebasco Services Incorporated
Excavation	Max J. Kuney
General Contractor	Robt J. Gobel
Electrical Sub	Power City Electric
Mechanical Sub	Spooner Mechanical
Landscape Architect	David Evans Assoc.

Following is a general schedule of project development:

Project Schedule and Duration of Tasks

Activity	Dates	Duration
Preliminary Engineering	January 10 - June 30, 1989	6 months
FERC License Amendment	June 30 - September 30, 1989	3 months
Turbine-Generator Procurement		
Bid Document	August-September 1989	2 months
Bidding	October - December, 1989	4 months
Notice to Proceed	February 1, 1990	
Final Design	March - August 31, 1990	6 months
Issue Cofferdam and Demolition Contracts	September - October 1990	2 months

Activity	Dates	Duration
Issue Powerhouse Excavation Contract	December 1, 1990 - February 14, 1991	2 months
Bidding for General Construction	October - December 1990	4 months
GCC Award	January 1991	
Notice to Proceed	February 14, 1991	
Construction Complete	June 1992	16 months

HYDROELECTRIC PLANT INSPECTION AND MAINTENANCE REQUIREMENTS

STANLEY J. HAYES, P.E. [1], and DAVID M. CLEMEN, P.E. [2]

Abstract

Many hydroelectric project owners have recently focused on their need for a comprehensive inspection and maintenance program. The need to develop this program is present in the industry due to aging hydro facilities which require greater maintenance; and due to the great increase in independent power producers in the last decade, which typically do not have an extensive base of hydro experience. Due to the great diversity of mechanical and electrical equipment associated with hydroelectric plants, many owners find a "uniform" maintenance and inspection schedule identifying maintenance time periods and the tasks to be performed for each major equipment item difficult to prepare. The authors recently produced a baseline maintenance and inspection document to assist small utilities or independent power producers in maintaining their facilities.

Introduction

Owners that are most in need of a comprehensive inspection and maintenance program fall into two broad categories. First, there is a large group of owners of older plants (35 years plus) that have significantly reduced their hydro maintenance staff and budgets during the last decade.

1) Stanley J. Hayes, Senior Mechanical Engr., R. W. Beck and Associates, 2101 Fourth Ave., Suite 600, Seattle, WA 98121-2375
2) David M. Clemen, Senior Electrical Engr., Harza Engineering Company, 233 S. Wacker Drive, Chicago, IL 60606-6392

Second, there is a sizable group of new owners, such as private developers, that have no in-house hydro maintenance expertise. Each group will have significantly different maintenance requirements for individual items of equipment, but the overall needs of both groups can be met to a great extent by applying a uniform maintenance and inspection program.

Significant aspects of the uniform maintenance and inspection program include: a) Categorization of equipment and components, b) Establishment of the maintenance management system, c) Development of detailed maintenance and inspection schedules, and d) Implementation of the program. Further definition of these significant aspects are as follows:

Categorization of Equipment and Components

Major Equipment Categories. Before developing any program, a system must be established to identify and track inspection and maintenance activities according to major equipment category and specific items. This identification system is absolutely critical; without it, there is no way to organize and analyze the data the program will produce. The major equipment categories for a hydroelectric plant were defined by the authors as listed below; categories may need to be added or deleted to suit specific owners' requirements:

- Hydraulic Turbines
- Governing Systems
- Generators and Excitation Systems
- Generator Circuit Breakers
- Oil-Filled Transformers
- Medium Voltage Switchgear
- Main Leads/Buswork/Surge Arresters
- Unit Control Switchboards and Control Equipment
- Low-Voltage Switchgear, Panelboards, Buses and Cables
- Motors
- Power Conduit/Steel Penstock and Draft Tubes
- Gates and Inlet Valves
- Cranes and Hoists
- Powerhouse Auxiliary Mechanical Equipment and Systems
- Powerhouse Auxiliary Electrical Equipment and Systems
- Powerhouse Buildings and Intakes
- Elevators

Component Designator System. Once the categories are established, a system of alphanumeric designators is needed to identify specific components or systems. The designator system must be sufficiently refined to provide individual designators for each item, component, or system for which discrete inspection and maintenance activities are required. The designator system developed by the authors includes fields for the following:

- Plant Name
- Unit Number
- Major Equipment Category
- Item, Component, or System
- Subdivision or Individual Element

Using the system, designator XYZ-01-TRB-RNR-BL2 would indicate Plant XYZ, Unit One, turbine, runner, blade number two. As another example, designator XYZ-03-GEN-HPL-P01 would indicate Plant XYZ, Unit Three, generator, high pressure oil lift system, pump number one. A master list of designator codes for each field can be readily developed to suit specific owners' requirements.

Maintenance Management System

Organization by Task Assignment. Before instituting an inspection and maintenance program, the system to organize and manage the activities must be firmly established. Since inspection and maintenance activities usually consist of individual, discrete tasks, any such program can be logically organized around task assignments. The individual task assignment sheet contains an inspection and testing check list required to be performed on the specific item of equipment. The task assignment sheet includes the following:

- A field for entering the date and signature of the inspector.

- A field for writing the status, reading, measurement, or test value.

- A blank field for comments or observations.

- Instructions for the disposition of the completed forms.

- Instructions for the action to take in the event of abnormal/emergency conditions.

- Maintenance procedures containing specific instructions for disassembly, servicing, tests, re-assembly, and start-up. The procedures include checklists and fields similar to those used for inspection and tests for recording measurements and test values, and observations.

- Tag-out procedures detailing specific instructions and steps for removing equipment or systems from service, and instructions and steps for returning equipment or systems to service.

Computer Maintenance Management System. Historically in the hydro industry, management of the task assignments was performed manually. Systems based on check sheets, card files, and/or folders were sufficient to allow owners to perform routine activities. However, such systems are not adequate to provide the detailed, readily-retrievable analytic data needed today to cost-effectively manage hydroelectric resources. The recent development of Computer Maintenance Management Systems (CMMS) offers hydro owners the sophisticated tools they need to manage maintenance activities in order to minimize both costs and unscheduled outages.

Centralized control of maintenance tasks, via a CMMS which generates task assignments, ensures a coordinated and consistent maintenance program for all categories (turbines/generators/gates, etc.) of equipment. The CMMS provides an orderly scheduling of task assignments for the total number of jobs required based upon the workers available and the time intervals involved. Furthermore, it is utilized to document equipment maintenance/repair histories, spare parts inventories for all equipment, and calculations of total maintenance costs in addition to cost trends for each individual equipment item. The CMMS also provides a database of all inspection and test data. Using the data, comparative and trending analyses can be performed so that major maintenance and repairs can be proactive rather than reactive. Several references for selecting and using a CMMS are listed at the end of this paper.

Maintenance and Inspection Schedules

Sources of Data. Maintenance and inspection schedules can be developed from a number of resources. Ideally, the starting point should be the equipment manufacturer's recommended inspection and maintenance schedules, obtained either from the equipment Operation and Maintenance Manuals (if available) or from the suppliers, if they or their successors

are still in business. Suppliers of similar equipment often provide useful guidelines. Owners of older units often find (much to their surprise!) that detailed schedules and instructions are available in their project files, but have been forgotten. Other project owners can be an excellent source of either equipment-specific or generic information. For example, an excellent maintenance guide for mechanical equipment is available from the USBR and is listed in the references. The uniform schedules developed by the authors drew from all the above-listed sources.

Uniform Schedules. Due to the fact that inspection, equipment maintenance, and testing are performed at different times, the uniform maintenance and inspection schedule identifies a specific time interval for each of these items. Naturally, the time periods or intervals specified can be adjusted for an individual owner based upon specific equipment problems, age of generators or turbines, etc., etc. Some partial extracts of the uniform maintenance and inspection schedule developed by the authors for "Hydraulic Turbine" and the "Generator and Excitation System" are shown in Tables 1 and 2.

Due to the vast array of types of generating equipment and auxiliary system components, it is especially important that the uniform maintenance schedule be "tailored" in these areas to the specific types and components installed in the individual power plant. Examples of specific considerations for hydraulic turbines and generators and excitation systems are given below to illustrate the criteria for "tailoring" the schedules to suit specific equipment.

Hydraulic Turbine Considerations. Shaft orientation considerations factor into the schedules; horizontal and vertical shaft bearings have different detailed maintenance requirements. The type of turbine runner is critical. For example, on Pelton turbine runners, one likely location for cracking is at the transition between the buckets and the hub disc. On Francis turbine runners, several typical zones of potential cavitation exist - at the inlet transition between the blades and the band, at the bottom of the low pressure side of the blade near the band, and other relatively well-defined areas. The inspection and maintenance schedules for the turbine runners should be tailored to focus on those areas known to be potential locations of cracking and cavitation, whether or not problems have been found in those areas on the specific units in question. Of course, an owner's specific "problem areas" must also be identified and highlighted in the schedules. Similar considerations are used to tailor the remainder of the turbine components.

Generator and Excitation System Considerations. The generator types range from the old asphaltic insulated multi-turn "chain" or "lap" type stator windings to the modern "Class F" insulated single turn bar stator

TABLE 1

UNIFORM INSPECTION AND MAINTENANCE SCHEDULE (PARTIAL)
HYDRAULIC TURBINE

ITEM	MAINTENANCE OR INSPECTION TASK	TASK INTERVAL	COMMENTS
Runner	Visual Inspection	Annual	
	Map cavitation, erosion, and/or cracking	Annual	
	Patch cavitated or eroded areas with epoxy	Annual	
	Weld repair cavitation, erosion, and/or cracking	Every 5 years	
Runner Wearing Rings	Visual inspection	Annual	
	Check clearance at four locations 90 degrees apart	Annual	
	Replace wearing rings	Wnen clearance is twice design	
Spiral Case/Stay Ring	Visual Inspection	Annual	
	Map erosion and/or cracking	Annual	
	Patch erosion with epoxy	Annual	
	Replace mandoor gasket	Annual	
	Clean drains and pressure taps	Annual	
	Touch-up paint	Annual	
	Weld repair erosion/cracking	Every 5 years	
	Measure spiral case/stay vane thickness	Every 5 years	
Wicket Gates	Visual Inspection	Annual	
	Record stem bearing clearances (where possible)	Annual	
	Record clearances at top, bottom, and between gates with gates in closed position	Annual	
	Operate gates through 3 cycles and observe in dry	Annual	
	Map cavitation and/or erosion	Annual	
	Patch with epoxy	Annual	
	Touch up paint	Annual	
	Weld repair	Every 5 years	

TABLE 2

UNIFORM INSPECTION AND MAINTENANCE SCHEDULE (PARTIAL)
GENERATOR AND EXCITATION SYSTEM

ITEM	MAINTENANCE OR INSPECTION TASK	TASK INTERVAL	COMMENTS
General	MEGGER TEST - stator winding, main and pilot exciter, main field winding, PMG housing, AC and DC oil pumps	Annual	
	PI TEST - stator winding	Annual	
	INSULATION PF TEST - stator winding	Every 2 years	
	Rotor Pole Drop Test	Every 2 years	
	Check the following:		
	Automatic brake operation, brake shoe wear	Annual	
	86G protective relay trips	Annual	
	Alarms as follows:		
	bearing temperatures	Annual	
	stator winding temperatures	Annual	
	air cooler temperatures	Annual	
	bearing oil levels	Annual	
	vibration levels	Annual	
	Visual examination of the following:		
	Structural members and fasteners	Annual	
	Foundation anchors and embedments	Annual	
	Stator core (front and back)	Annual	
	Stator windings	Annual	
	Main and pilot exciter windings	Annual	
	Field windings	Annual	
	Rotor pole connections	Annual	
	Main and neutral connections	Annual	
	Heat exchangers and cooling system piping	Annual	
	Excitation system, including the AVR, field rheostat, brushes, slip rings, and commutator(s)	Annual	
	High pressure oil lift system	Annual	

windings. The inspection and testing frequency of the insulation, for example, should be tailored to the specific insulation type and history of performance. The exciter components involved often range from the older style main and pilot exciters with field rheostats to the modern day static excitation systems. The maintenance and testing requirements are much more comprehensive for the older style excitation systems. Consequently, a "tailored" maintenance program is essential to adequately maintain the specific components involved.

Further "tailoring" of the uniform maintenance and inspection base schedules to suit the needs of the individual Owner would take into account specific experiences within the Owner's hydroplants, the number of testing and maintenance personnel available, and the extent of in-house expertise. The less expertise available within the Owner's organization, the greater the amount of specialized maintenance tasks that must be subcontracted. However, with identification and tracking of the individual maintenance and inspection tasks on an annual basis, the manpower and budget required becomes more manageable.

Implementation

Commitment. A common mistake that occurs when introducing new maintenance programs is to underestimate the level of commitment required to implement the program, and the amount of personnel re-training or education required to effectively execute the program. A new maintenance program requires a total management commitment over a number of years (approximately 3 - 5 years), in addition to active employee participation. In the short term, this will result in increased capital expenditures for the CMMS hardware and software and additional maintenance and testing equipment. "Soft" costs for personnel will also increase - additional personnel may be required to put the program in place, and all employees will need to be trained in the program. Owners must remain focused on the long-term benefits: reduced long-term maintenance costs and increased energy production resulting from fewer unscheduled outages.

Adjustments to Program. After the maintenance program is successfully implemented, it should be periodically re-evaluated and adjusted. Evaluations one year, three years, and five years after implementation are suggested, but other intervals may be more appropriate as circumstances dictate. One major benefit of the program is the detailed, organized data it provides. Owners should use this information to fine-tune the program; provided the program was well-designed, any adjustments should be relatively minor and can be implemented at low cost. The result will be a sound program optimized for a particular Owner's needs.

Summary

A "tailored" maintenance and inspection program which identifies all essential tasks for a hydro plant and, in addition, takes into account the personnel/level of expertise of the organization is a valuable tool. It will allow an organization to optimize the tasks for the specific plants; schedule maintenance events in an orderly fashion; and identify and track tasks/costs required to be performed internally and by outside contractors. Formation of a comprehensive program requires four major phases: detailed categorization of all equipment and systems, establishment of a computerized maintenance management system, development of site-specific and component-specific maintenance and inspection schedules, and implementation of the program, consisting of initiation of the program, operation of the program, and subsequent fine-tuning.

The categorization system and uniform maintenance and inspection schedule recently developed by the authors fills the need for those agencies/ utilities / owners that do not have extensive manpower or a significant amount of hydro expertise present. Together with a suitable CMMS and a committed implementation plan, they provide a comprehensive maintenance and testing program that provides owners with essential tools to efficiently and effectively manage their hydroelectric resources.

Selected References

1. Terry Wireman, Computerized Maintenance Management Systems, Industrial Press, Inc.

2. Thomas Marketing Information Center, 1991 CMMS Directory and Comparison Guide and CMMS Users Handbook, Thomas Publishing Co.

3. United States Department of the Interior, Bureau of Reclamation, Water Operation and Maintenance Bulletin No. 160, Maintenance and Periodic Inspection of Mechanical Equipment at Hydroelectric Plants.

Use of Hazard Analysis in Maintenance

Walter O. Wunderlich, M. ASCE[1]

Abstract

Hazard analysis can provide a planning framework for maintenance and rehabilitation activities. Hazards can increase the probability of harmful events, their magnitude and consequences. Thereby, hazards can affect safety, economy and reliability of systems. The evaluation of risky decisions is an important tool for prioritizing hazard control action. Quantitative methods for hazard analysis are discussed using an example.

Introduction

The true state of hydraulic systems is often obscured by limited access and limited measuring opportunities. Hidden or unrecognized hazards can trigger event sequences that may lead to component or system failure. Decisions whether or not to control hazards through maintenance or rehabilitation can be risky, as there is uncertainty about the true system state, i.e., about hazards in the form of possible failure modes that may be triggered by unusual loadings imposed by chance events. Decision errors may have costly consequences including needless shutdowns, unexpected malfunctioning, maintenance and rehabilitation cost overruns, and system failure.

In recent years, increasing safety concerns in various infrastructure sectors, such as hazardous waste disposal (Cheremisinoff, 1992), nuclear reactor safety and dam safety (NRC, 1983), have led to an increased use of hazard analysis methods. Such probabilistic methods have definitely also applications in hydraulic infrastructure maintenance planning (Wunderlich, 1991). Here an outline of hazard analysis is given and an example is discussed.

[1]Consulting Engineer, 3221 Essary Dr., Knoxville, TN 37918.

Hazard and Risk

A hazard is a condition which introduces or increases the probability of loss from a (chance) event, be it by making the event more likely or by increasing its consequences. A structural inadequacy is a hazard that increases the probability of failure and makes the exposure to harm or loss by a structural failure more likely. Risk is the exposure of persons and/or property to chance events that may cause personal harm or property loss. Risk may be measured by objective statistical measures (objective risk) or by individual measures based on the risk perception of risk takers (subjective risk). The essential constituents of risk are (chance) event, exposure and probability. All are affected in various ways by hazards. Risk always exists if there is a probable exposure to harm or loss.

An approach that is sensitive to risk must deal with the chance outcomes of decisions. Comparing alternatives on the basis of 'expected values' (EV) is not sensitive to risk. Suppose there are two alternatives. The first has two 50/50 chance outcomes, 8 (gain) and -4 (loss), and the second has two 50/50 chance outcomes, 4 and 0. Both alternatives have EV = 2, but they are certainly not the same, as far as exposure to possible loss, i.e., risk, is concerned. An objective risk measure, the variance, indeed shows that the first alternative has a variance of 36, while the second has a variance of 4, in other words, the first alternative is more risky.

Maintenance as an Intervention Process

Maintenance is primarily concerned with detection and control of internal hazards, conditions that can trigger internal chance events, such as foundation failure, cracks in vital components, bearing failure in machinery, gate failure, trashrack collapse, etc. Internal chance events may start out as undetectable flaws. If they remain undetected, or unattended, the event sequences may evolve and lead to unexpected component outage or system failure. Maintenance can be construed as an intervention process that interferes with a continually ongoing stochastic deterioration process, a process whose states are not entirely predictable. The basic steps of the intervention process are :

1. surveillance (hazard detection)
2. event and consequence identification (event trees)
3. system analysis (failure mode and damage analysis)
4. decision analysis (evaluation and ranking of alternatives)
5. decision finding and corrective action (hazard control).

Surveillance detects incipient flaws and triggers the intervention sequence. Here only decision analysis dealing with the evaluation of various corrective action alternatives will be discussed.

Decision Tree

If a decision must be made, such as repairing or rehabilitating a structure, there are usually several alternatives by which the job can be done. All decision alternatives make up the decision tree (Keeney and Raiffa, 1976). An example with five such alternatives is shown in Figure 1. The data used here were taken from a Bureau of Reclamation manual (ACER, 1986). At the decision point, marked by the square, the decision maker (DM) has the choice of several alternatives. Only the branch "No action" is shown in detail. If chosen, the DM (or those for whom he decides) incurs exposure to chance events (marked by circles) over which he has no or only limited control. Uncontrollable chance events are natural events, such as floods and earthquakes. Partly controllable chance events are those that may be triggered by various structural hazards, such as identifiable possible failure modes under natural events.

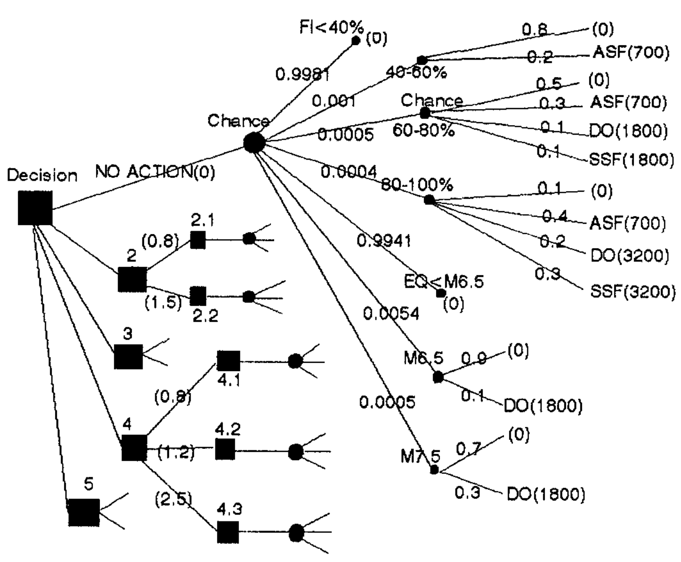


Figure 1 Decision Flow Diagram with Decision and Chance Branchings (after data from ACER, 1986).

The following structural failure modes are identified: dam overtopping (DO), auxiliary spillway failure (ASF), and service spillway failure (SSF). The numbers in parentheses at the end of the chance branches represent the exposures (consequences), here damages in $million. The floods, given in percentages of the probable maximum flood (PMF), can be safely passed as long as they are less than the 40%-PMF. Also, the dam can safely resist earthquakes less than M6.5.

The natural events include two sets of events that can occur independently during the time step: floods and earthquakes, each of them having event probabilities, $P[\theta]$, that sum to 1. The failure modes have probabilities conditioned on the natural events, $P[\phi|\theta]$. A summary of these probabilities for the <u>status quo</u> and nine hazard control alternatives is given in Table 1.

Table 1 Failure Mode Probabilities of Alternatives (ACER, 1986)

Events: Floods/ Earthqu.	Failure Mode	Failure Mode Probabilities for Alternatives									
		Status quo	Alt. 2.1	Alt. 2.2	Alt. 3.1	Alt. 3.2	Alt. 4.1	Alt. 4.2	Alt. 4.3	Alt. 5.1	Alt. 5.2
1	2	3	4	5	6	7	8	9	10	11	12
0- 40%P	None	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
40-60%P	None	0.80	0.80	0.80	0.80	0.80	1.00	1.00	1.00	0.80	0.80
	ASF	0.20	0.20	0.20	0.20	0.20	0.00	0.00	0.00	0.20	0.20
60-80%P	None	0.50	0.60	0.60	0.60	0.60	0.50	0.92	0.95	0.60	0.60
	ASF	0.30	0.30	0.30	0.30	0.30	0.30	0.00	0.00	0.30	0.30
	DO	0.10	0.00	0.00	0.00	0.00	0.10	0.08	0.05	0.00	0.00
	SSF	0.10	0.10	0.10	0.10	0.10	0.10	0.00	0.00	0.10	0.10
80-100%	None	0.10	0.20	0.30	0.20	0.30	0.10	0.20	0.75	0.20	0.30
	ASF	0.40	0.40	0.40	0.40	0.40	0.40	0.30	0.00	0.40	0.40
	DO	0.20	0.10	0.00	0.10	0.00	0.20	0.20	0.10	0.10	0.00
	SSF	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.15	0.30	0.30
< M6.5	None	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
M6.5-7	None	0.90	0.95	1.00	0.95	1.00	0.90	0.92	0.95	0.95	1.00
	DO	0.10	0.05	0.00	0.05	0.00	0.10	0.08	0.05	0.05	0.00
M7.5	None	0.70	0.85	0.95	0.85	0.95	0.70	0.70	0.80	0.85	0.95
	DO	0.30	0.15	0.05	0.15	0.05	0.30	0.30	0.20	0.15	0.05

Figure 1 also depicts alternatives that are executable in substeps: Alternative 2.1 proposes raising the dam to eliminate the likelihood of overtopping for the 80%-PMF, and Alternative 2.2 eliminates the likelihood of overtopping for the 100%-PMF, etc. The numbers in parentheses on decision branches are costs incurred by these alternatives.

Decision Tree Evaluation

The decision tree in form of a computation table to carry out the analysis is shown in part in Table 2. The

first six columns contain input information. Col. 3 gives the flood and earthquake event probabilities, P[θ], and col. 5 gives the failure mode probabilities, P[ϕ|θ]. The expected damage cost for alternative k is calculated by

$$D_k = \sum_j \{(P[\theta_j] \ (\sum_i D_i \ P[\phi_i | \theta_j]))\} \tag{1}$$

The inner sum calculates the expected damages for each failure mode i under event j, and the outer sum cumulates these expected damages per event, D_j, over all natural events j, and obtains the total expected damage of the alternative, Dk. For the No-action alternative, k = 1, the total expected damage cost from four floods events and three earthquakes (both including zero-damage events) is $D_1 = D_{f1} + D_{e1}$, with indexes f and e referring to floods and earthquakes, respectively: D_{1f} = 0.9981*0 + 0.001*140 + 0.0005*570 + 0.0004*1880 = 1.18, and D_{e1} = 0.9941*0 + 0.0054*180 + 0.0005*540 = 1.24, hence D_1 = 1.18 + 1.24 = 2.42 (in $million/yr), shown in Table 2, bottom of col. 7.

Table 2 Evaluation of Decision Alternatives

Action Alternat.	Events: Floods/ Earthqu.	Probability of Floods/ Earthquakes	Failure Mode	Failure Mode Probabil.	Damage Cost $M	Probable Dam. Cost $M/yr
1	2	3	4	5	6	7
Alt. 1	0-40%PM	0.9981	None	1.00	0	0.000
Status	40-60%P	0.0010	None	0.80	0	0.000
quo		0.0010	ASF	0.20	700	0.140
	60-80%P	0.0005	None	0.50	0	0.000
		0.0005	ASF	0.30	700	0.105
		0.0005	DO	0.10	1800	0.090
		0.0005	SSF	0.10	1800	0.090
	80-100%	0.0004	None	0.10	0	0.000
		0.0004	ASF	0.40	700	0.112
		0.0004	DO	0.20	3200	0.256
		0.0004	SSF	0.30	3200	0.384
	<M6.5	0.9941	None	1.00	0	0.000
	M6.5-7	0.0054	None	0.90	0	0.000
		0.0054	DO	0.10	1800	0.972
	M7.5	0.0005	None	0.70	0	0.000
		0.0005	DO	0.30	1800	0.270
Summary 1		Reliability:	Fld's & EQ's	0.9992	0.9993	2.419

The reductions in expected damages (EV's) are construed as benefits. The net benefit, NB, and the benefit/-cost ratio, BC, for each alternative are calculated by:

$$NB_k = dD_{k,0} - C_k - BL_k \tag{2}$$

and

$$BC_k = dD_{k,0}/(C_k + BL_k) \tag{3}$$

NB_k is the net benefit of alternative k, $dD_{k,0}$ is the difference in expected damage between the No-Action alternative (status quo) and alternative k, C_k is the cost of rehabilitation, BL_k is a benefit loss for non-structural measures, such as operating at lower water levels. The results of these calculations are summarized in columns 9 and 10 of Table 3 under 'EV-based' NB and BC.

Table 3 Summary of Decision Criteria

Decision Alterat.	CE $M	EV $M	RP $M	Cost of Rehab $M	Benefit Loss $M	NB (CE) $M	BC (CE) 1	NB (EV) $M	BC (EV) 1
1	2	3	4	5	6	7	8	9	10
0	9.19	2.42	6.77	0.00	0.00	0.00	-	-	-
2.1	6.27	1.58	4.69	0.82	0.00	2.11	3.58	0.02	1.03
2.2	3.71	0.88	2.84	1.55	0.00	3.93	3.53	-0.01	0.99
3.1	6.27	1.58	4.69	0.00	0.33	2.60	8.95	0.51	2.57
3.2	3.71	0.88	2.84	0.00	0.57	4.91	9.58	0.97	2.70
4.1	8.99	2.28	6.71	0.82	0.00	-0.62	0.25	-0.68	0.17
4.2	7.95	1.84	6.11	1.23	0.00	0.01	1.01	-0.65	0.47
4.3	4.34	1.03	3.31	2.45	0.00	2.40	1.98	-1.06	0.57
5.1	6.27	1.58	4.69	0.41	0.16	2.35	5.12	0.27	1.47
5.2	3.71	0.88	2.84	0.82	0.29	4.37	4.97	0.44	1.40

$M means "million dollars/year"
"1" means a non-dimensional number.
(EV) means "based on expected value."
(CE) means "based on certainty equivalent."

Risk Analysis

The problem with expected values is that two decisions with the same expected value can pose quite different risks, as was already pointed out. Risk analysis must analyze the acceptability of the individual outcomes of the alternatives.

Objective risk assessment looks at objective information on departures from the mean. The computation of D_{fl} and D_{el} above shows that the No-action damage cost is made up of expectations for each event, ranging from 0 to 1880 for floods, and from 0 to 540 (all in $million) for earthquakes. Furthermore, depending on the failure mode, damages range from 0 to 3200 ($million). A measure for the depar-

tures from the mean is the variance. It is computed by substituting $(x_i - EV')^2$ for D_i in Eq. 1 to produce Var_k for each alternative, with x_i being the damage costs in col. 6 of Table 2, and EV' being the annual expected damage shown at the bottom of col. 7, but capitalized (here by a factor 12) to make it compatible in units with x_i. Table 4, col. 4 summarizes these results.

Another objective risk criterion is the overall reliability of an alternative, i.e., the probability of no failure. It is calculated by adding the probabilities of all no-failure events. For alternative 1, the reliability of no flood damage is: $R_{f1} = 0.9981 + 0.001*0.8 + 0.0005*0.5 + 0.0004*0.1 = 0.9992$. Flood and earthquake reliabilities are summarized in Table 4. Total reliability would require that neither damaging floods nor damaging earthquakes occur.

<u>Subjective risk</u> assessment makes use of personalized measures of attitude toward risk by those who are exposed to harm. The expected value, as was already pointed out, does not assess risk, but it is a guiding value: the risk-averse DM perceives a larger, and the risk-prone DM perceives a smaller damage than the EV. Therefore, the EV is a risk-neutral value (Keeney and Raiffa, 1976).

Table 4 Ranking of Alternatives

Decision Alterat.	Reliabil. Floods 1	Reliabil. Earthqu. 1	Variance ($M)^2	Ranking by NB (EV)	Ranking by BC (EV)	Ranking by NB (CE)	Ranking by BC (CE)	Ranking by RP (CE)
1	2	3	4	5	6	8	9	10
0	0.99919	0.99931	6402	-	-	-	-	6
2.1	0.99928	0.99966	3827	5	6	7	5	3
2.2	0.99932	0.99998	1924	6	5	3	6	1
3.1	0.99928	0.99966	3827	4	2	4	2	3
3.2	0.99932	0.99998	1924	1	1	1	1	1
4.1	0.99939	0.99931	6131	8	9	9	9	5
4.2	0.99964	0.99942	5019	7	8	8	8	4
4.3	0.99988	0.99963	2584	9	7	5	7	2
5.1	0.99928	0.99966	3827	3	3	6	3	3
5.2	0.99932	0.99998	1924	2	4	2	4	1

$M means "million dollars/year"

"1" means a non-dimensional number.

(EV) means "EV-based"

The personal attitude of a DM toward risk can be elicited by asking him a set of questions that probe his willingness for accepting risky choices. The DM can also directly 'shape' a utility function so that it represents his attitude toward risk (Keeney and Raiffa, 1976). For brevity we assume the latter approach here. Such utility

functions, due to their arbitrariness, may produce very different 'utility' weights. But the essential characteristic of the adopted function is its shape, which must display 'risk-aversion,' in other words, it must produce larger than expected damages. The utility function adopted here and displayed in Figure 2 is

$$u(x)= -\exp(0.001x) \tag{4}$$

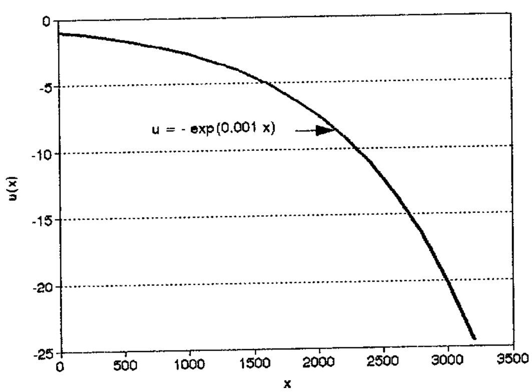


Figure 2 Decreasing Utility Function for Risk-averse Decision Maker.

By use of Eq. 4, the damages, x, in col. 6 of Table 2 are converted into utilities, u. By substituting the u's into Eq. 1, expected utilities are calculated for the flood and earthquakes separately by extending the sums only over the respective event outcomes. The expected utilities are then reconverted by the inverse,

$$x = \ln(-u)/0.0025 \tag{5}$$

into damage costs ($million/yr) and added to form the 'certainty equivalents,' CE, a 'value for certain' in lieu of the risky outcomes of the alternatives. The CE is the expected damage cost seen by the risk-averse DM. The characteristic of a 'decreasing' risk-averse utility function (u decreases as damage increases) is expressed by

$$RP = CE - EV > 0 \tag{6}$$

with RP being the risk premium. The risk premium is the 'surcharge' that the risk-averse DM adds to the expectation. The smaller the risk premium, the less risk he perceives. The benefit/cost analysis proceeds from here with using CE instead of EV in NB and BC calculations.

Discussion of Results

The decision criteria for nine alternatives and the status quo (Alternative 1) and their rankings are shown in Table 3 and 4. The costs and benefits for the substeps are cumulated to reflect full costs and benefits for the sublevels. The criteria and their preferred levels (in parentheses) are as follows: NB (high), BC (high), Reliability (high), RP (low), and Variance (low). Assigning a 1 to the most preferred level and successively higher numbers to less preferred ones, giving the same numbers to ties, produces the rankings shown in Table 4. While the rank number sequences under each criterion are different, less preferred groups of alternatives (high numbers) can be easily spotted, such as alternative 4 and its substeps. Alternative 3.2 stands out as the most preferred one. Comparing the CE-based with the EV-based rankings does not reveal major differences. This result was unexpected. It may be due to the limitations of the example, but ranking aspects could not be pursued here.

The CE-based NB's and BC's are 'inflated' by higher risk-costs. Correspondingly, also the risk cost reductions, $dD_{k,0}$, are higher for the risk-averse DM, while they are relatively small EV-differences for the risk-neutral DM.

The high probabilities of no-failure, expressed by the reliabilities, even for the no-action alternative, in Table 4, col. 2 and 3, are the complements of the low probabilities of major natural events. As the probability of the chance event approaches zero, the probability of exposure and, thus, risk vanishes, and no further damage reduction is possible. If an especially risk-averse (or otherwise motivated) DM wants to give such investments high 'utility', he can shape his utility function so that it is sensitized to large damages. Then, also large damage reduction benefits appear, which justify large capital investments. This trend to increased risk reduction benefits is demonstrated by the differences in CE- and EV-based NB's in Table 3, col. 7 and 9, and the BC's, col. 8 and 10. The high values of the CE-based criteria show that the investment costs are relatively small compared to the 'inflated' damage reductions, $dD_{k,0}$. The utility-based benefits (damage reductions) never materialize in tangible monetary form, in contrast to the rehabilitation costs that must be procured from some source. But the 'inflated' benefits can possibly help guide the decision process toward alternatives of some identified 'utility.' Here it is risk reduction.

Conclusions

Hazard analysis is described as a framework for maintenance. Hazard detection and control are important activi-

ties of any preventive maintenance program. Decision analysis is important, as it can guide investment toward the most effective hazard control alternatives. Decision analysis in the face of uncertainty must judge alternatives by their chance outcomes. Objective and subjective risk measures for this purpose were discussed. The risk-conscious decision maker should use all available information and various criteria to view the economic and risk aspects of alternatives from as many perspectives as possible. The risk-averse decision maker perceives a higher (or a more negative value) than the expected damage, while the risk-prone decision maker perceives a lower one. Therefore, the expected value (EV) is actually risk-neutral. Adding the risk premium to the EV produces the certainty equivalent, CE, the risk-averse decision maker's 'surcharged' cost or damage expectation. In the example, a utility function was used that produced CE's about four times larger than the EV's (Table 3). The decision alternatives were ranked using CE- and EV-based criteria. No major difference in ranking was found. The reasons could not be pursued. The outlined risk assessment approaches are deemed to have potential for more extensive problems, specifically for multi-utility problems.

References

ACER (1986). "Guidelines to Decision Analysis." ACER Technical Memorandum No. 7, Bureau of Reclamat., Denver.

Cheremisinoff, P.N. (1992). "Hazard Analysis for Waste Reduction." The National Environ. Journ., March/April 1992.

Keeney, R. L. and H. Raiffa (1976). Decisions with Multiple Objectives: Preferences and Value Tradeoffs. J. Wiley & Sons, New York.

NRC (1983). Safety of Existing Dams: Evaluation and Improvement. Committee on the Safety of Existing Dams, National Research Council, National Academic Press, Washington, D.C.

Wunderlich, W. O. (1991). "Probabilistic Methods for Maintenance." Journ. of Eng. Mechanics, ASCE, 117:9, 2065-2078.

CAVITATION REPAIR BY WELDING WITHOUT GOUGING

Avaral S. Rao[1]
Ken S. Hawthorne[2]

Abstract

Welding procedures used to overlay cavitation damage of hydraulic turbines without gouging or grinding the cavitated areas were developed through an empirical analysis. Results of weld overlay experiments on the cavitated test specimens and subsequent field tests showed that Pulse Gas Metal Arc Welding procedures can be applied for overlaying directly on the cavitated areas where the depth of cavitation damage is less than 1.5 mm.

Introduction

Cavitation damage on hydraulic turbines is one of the more costly maintenance problems for hydraulic utilities. A survey (1), sponsored by the Canadian Electrical Association has estimated that the annual repair due to cavitation erosion on turbines in Canada costs nearly $38 million in labour and lost capacity. A similar survey by the Electrical Power Research Institute (EPRI) has concluded (2) that approximately 80% of older hydraulic electric and pump storage turbines in the U.S has suffered cavitation erosion in one degree or another. Thus the economic benefit due to the reduction of cavitation in hydraulic turbine can be spectacular.

The cavitated areas of the hydraulic turbines are periodically repaired to restore the original profile of the turbine. In majority of the cases, the cavitated areas are repaired in-situ for economic reasons. Overlay welding

[1] Senior Materials Engineer, Powertech Labs Inc., 12388 88th Avenue, Surrey, B.C., Canada, V3W 7R7

[2] Engineering Supervisor, Apparatus Dept.,BC Hydro, 6911 Southpoint Drive, Burnaby, B.C., Canada, V3N 4X8

has been the most popular method to refurbish cavitated areas on hydraulic turbines.

The first step for welding repair, namely weld preparation, can be accomplished using carbon air arc gouging, plasma arc gouging or grinding methods. These weld preparation methods produce fumes and dust which can be easily extracted in a shop floor using fume extractors. During cavitation repair in the field, it is very difficult to remove all the dust and fume areas of cavitation repair. Therefore, welders are forced to wear cumbersome breathing air masks.

During the early stages of cavitation (frosting) on hydraulic turbines, the damage is limited to a few millimetres. In such situations, the preparation of the cavitated surface either by grinding or by arc gouging may not be required. Instead, cavitation resistant weld overlay can be directly deposited on this surface. However, conventional welding procedures may not provide a sound weld under this condition. It would require specialized welding procedures to provide good fusion between the cavitated surface and the weld metal.

The overall objective of this study has been to develop a welding procedure which can be used to repair cavitation damage on hydraulic turbines without gouging or grinding the cavitated area. In order to meet this objective, the study was divided into the following tasks:

(a) Reproduction of cavitation damage on test specimens to be used in the weld overlay experiments
(b) Development of a welding procedure for overlay on the cavitated area
(c) Testing and evaluation of weld overlays to establish acceptance criteria
(d) Field test

Experimental Procedure

Materials

The following four base materials were used in this study.

(a) The cast martensitic stainless steel plates (ASTM A 743 Grade CA6NM).
(b) Carbon steel plates (ASTM A 36) overlaid with an austenitic stainless weld metal (308).
(c) Carbon steel plates (ASTM A 36) overlaid with a cobalt based weld metal (Stellite 21)[3].
(d) Carbon steel plates (ASTM A 36) overlaid with a low cobalt containing austenitic steel weld metal (Hydroloy HQ 913)[3].

---

[3] Stellite 21 and Hydroloy HQ 913 are trademarks of Stoody Deloro Stellite, Inc.

The cast martensitic stainless steel plate was 175 mm long, 125 mm wide and 25 mm thick. Three sets of carbon steel plates were used as the base material for the overlay. Each of these plates was approximately 300 mm long, 200 mm wide and 25 mm thick. Three layers of the above mentioned weld metals were deposited on the top surface of the carbon steel plates using well established pulse gas metal arc welding procedures. The weld overlay was ground using a surface grinder. The ground surface was inspected for weld defects using a fluorescent dye penetrant examination method.

Cavitation Damage on test specimen

A cavitating fluid jet (Cavijet) method was used to create the cavitation damage on the test specimens. The detailed development of this experimental procedure is described in reference 3. A central body nozzle of 69 MPa (10,000 psi) was used to produce fine cavities in the jet stream. The jet stream was directed toward the surface of the test specimen in the immersed condition. Several traverses along the X-axis were made to obtain a cavitated area of 150 mm long and 12 mm wide. Three different depths of cavitation damage were produced on martensitic stainless steel (CA6NM) and 308 austenitic stainless steel weld overlay test specimens. Cavitation damage was limited to one location on the Stellite 21 and Hydroloy HQ 913 weld overlay test specimens.

Overlay welding on cavitated surface

Experiments of welding directly on the cavitated areas were conducted in two stages. In the first stage, a set of Pulse Gas Metal Arc Welding procedures was evaluated by welding on the ground surface of the test specimens. The influence of filler metal size, shielding gas and tie-in(overlap) on dilution and wettability of the weld metal were evaluated for optimization. The detailed analysis of this optimization is explained in reference 3. The optimized welding procedures from that study are listed in Table 1. In the second stage, the optimized welding procedures were tested by welding directly on the cavitated surface of test specimens in horizontal and vertical positions. A 450 amps transistorized invertor power source welding machine was used in this study. Two to four seams of weld bead were deposited on the cavitated surface of the specimen. Photographs of the weldments were taken using a 35 mm camera.

Non destructive examination

The integrity of the weldment was evaluated using ultrasonic and fluorescent dye penetrant examination methods.

Table 1
Optimized Welding Parameters for Overlay Welding

(a) - For Horizontal Applications (2G)

Substrate	Electrode Type	Electrode Diameter (mm)	Average Current (Amps)	Arc Voltage (Volts)	Wire Feed Speed (IPM)	Approx. Travel Speed (IPM)	Frequency (Hz)	Pulse Width (ms)
CA6NM	ER308-16	.9	110	21.5	246	8-10	86.3	4.0
CA6NM	ER308-16	1.2	150	23.0	167	8-10	94.6	3.0
308 Overlay	ER308-16	.9	95	19.5	195	8-10	69.9	4.0
308 Overlay	ER308-16	1.2	145	21.0	167	8-10	94.6	3.0
Stellite 21 Overlay	Stellite 21	1.2	120	17.0	158	8-10	70.0	2.5
HQI Overlay	HQ-913	1.6	125	18.0	105	8-10	51.0	3.75

(b) For Vertical Applications (3G)

Substrate	Electrode Type	Electrode Diameter (mm)	Average Current (Amps)	Arc Voltage (Volts)	Wire Feed Speed (IPM)	Approx. Travel Speed (IPM)	Frequency (Hz)	Pulse Width (ms)
CA6NM	ER308-16	.9	90	21.5	216	4-7	69.9	4.0
CA6NM	ER308-16	1.2	125	21.0	137	4-7	88.0	2.9
308 Overlay	ER308-16	.9	80	18.0	129	4-7	54.1	4.0
308 Overlay	ER308-16	1.2	110	18.5	146	4-7	88.0	2.9
Stellite 21 Overlay	Stellite 21	1.2	100	17.0	120	4-7	45.0	2.5
HQI Overlay	HQ-913	1.6	110	17.5	80	4-7	36.5	3.75

Destructive tests

Subsequent to the non destructive examination, the weldments were sectioned along the transverse direction of welding. Metallographic samples of the weldments were taken from each weldment. These samples were ground, polished and etched to reveal the microstructure of weld metal, fusion zone and the heat affected zone of the weldment. An optical microscope was used to examine the weldment. Macrographs of the weldments were taken using a 35 mm camera with the macro lens attachment. Microhardness scans of the weldments were carried out using a Tukon Microhardness Tester.

Results and discussion

Cavitation test specimens

The machined surface of the martensitic stainless steel (CA6NM) plates and the ground surface of carbon steel plates containing 308 austenitic stainless steel weld overlay did not indicate defects upon fluorescent dye penetrant examination. The ground surface of the Stellite 21 and Hydroloy HQ 913 weld overlays showed small amounts of porosity. Microfissuring was also observed on the surface of the Stellite 21 overlay.

The cavitated areas on the 308 weld overlay test specimen are shown in Figure 1. The dimensions of the cavitated surface on test specimens are shown in Table 2. Very little damage was seen on the Hydroloy HQ 913 weld overlay test specimen after 30 hours of cavitation under the cavitating fluid jet. Sand blasting and cross grinding with the coarse grit grinding wheels were tried to simulate cavitation damage on Hydroloy HQ 913 test specimen. The damaged surface on the Hydroloy HQ 913 weld overlay due to cross grinding with the coarse grit was closer to that of the cavitating fluid jet. Therefore cross grinding was used to prepare the Hydroloy HQ 913 specimen. The cavitated surface on the test specimens is similar to the early stages of cavitation erosion (known as frosting) on hydraulic turbines. Thus it clearly indicates that the cavitating fluid jet can be used to simulate cavitation damage on hydraulic turbines.

Cavitation repair weldments

Figure 2 shows the photographs of typical weldments. All weldments appeared sound to the naked eye.

Ultrasonic examination of the weld overlays gave inconsistent results and all the weld overlays showed some amount of porosity. However, a slightly higher amount of porosity was observed on the Hydroloy HQ 913

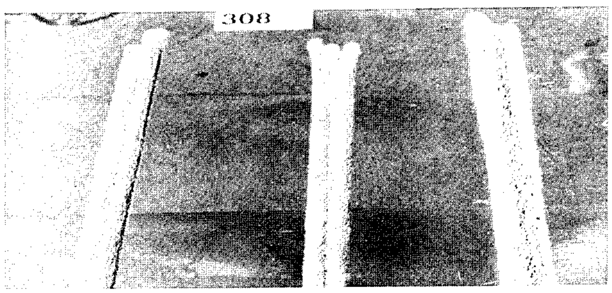


Figure 1 Cavitated areas on 308 weld overlay

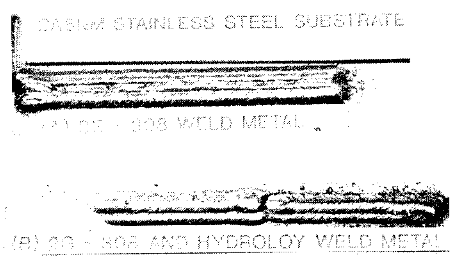


Figure 2 Weld overlay on cavitated areas of CA6NM
(A) 308 weld metal in 3G position (B) 308 (left) and Hydroloy (right) weld metal in 2G position

and Stellite 21 weld overlays. It is suspected that the type of wire used is responsible for the excess porosity. A hard drawn solid wire of 308 stainless steel was used as the filler metal whereas a cored type wire was used in the Hydroloy and Stellite 21 wire. Fluorescent dye examination of weldments failed to show any defects on the surface of the weldment.

Typical macrographs of the weld overlay are shown in Figure 3. A good fusion of the weld metal on the base metal was seen in all weld overlays. The weld overlay on CA6NM and 308 didn't show either lack of fusion defects or porosity. But the weld overlays on Hydroloy and Stellite 21 weld metal showed some porosity in the weld metal.

Table 2

Dimensions of the Cavitated Areas

No.	Test Specimen	Length (mm)	Width (mm)	Depth (mm)
1	308	150	12	0.76
2	308	150	12	1.52
3	308	150	12	3.0
1	CA6NM	150	12	0.76
2	CA6NM	150	12	1.52
3	CA6NM	150	12	3.0
1	Stellite 21	150	12	3.0
1	Hydroloy HQ 913	150	3	0.25

(a) (b)

Figure 3 Macrograph (A) CA6NM/308 weldment
(B) 308/308 weldment (mag. 3.5X)

The microhardness profiles along the heat affected zone of the weldments did not show any abrupt change except in martensitic stainless steel(CA6NM) weldments. The microhardness on the heat affected zone of the martensitic stainless steel showed higher hardness than that of the unaffected base metal. Similar results were observed in the earlier study (4) and it was found that the higher hardness is not detrimental to the integrity of the weld overlay.

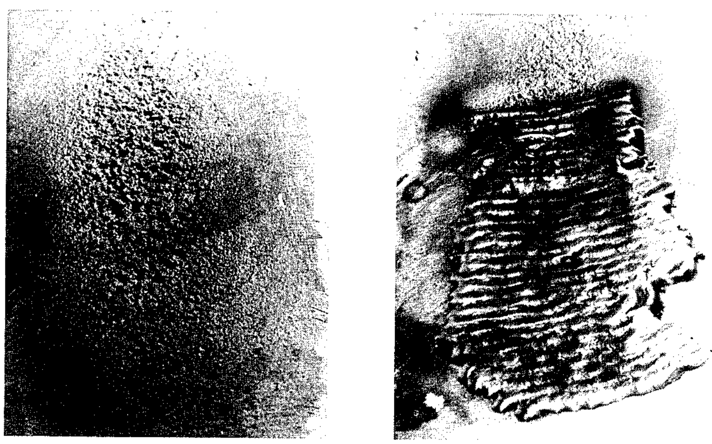
Field Test

As the final part of this project, a field evaluation of the welding procedures has been conducted. The object of the field test was to verify the laboratory findings. A Francis type hydraulic turbine runner, made of cast martensitic stainless steel (ASTM A 743 grade CA6NM), was chosen for the field test. Historically, this runner showed shallow cavitation damage over a large area on the suction side of the blade. This feature is attractive for carrying out the overlay welding on the cavitated surface without excavating the damaged area.

Three sections of the band (between blades 2 and 3, 10 and 11, 12 and 13) were selected for the field test. Each of these areas shaped like an ellipse of 165 mm major axis and 65 mm minor axis. The depth of the cavitation damage on these areas was measured using a dial gauge with a sharp tip(Pit gauge). The centre of cavitation damages was approximately 3 mm deep whereas the peripheral areas were shallower than 3 mm.

The cavitated areas were divided into two sections and one weld metal was chosen on an individual section. Prior to welding, the surface to be repaired was thoroughly cleaned using a stainless steel brush and subsequently it was preheated to 100°C using a propane torch. The preheating was necessary to get rid of moisture on the surface to be overlaid. A layer of the weld metal was deposited on the surface using the Pulse Gas Metal Arc Welding procedures shown in Table 1. The overlay welding was done in the horizontal position(2G). When the welding was completed, the overlay was ground smooth to the original contour of the band. The ground surface was inspected by visual examination followed by fluorescent dye penetrant examination methods. An attempt was made to use an ultrasonic examination method; however meaningful data could not be obtained because of the shallow weld and the coarse structure of the casting.

Figure 4 shows a typical weld overlay repair. Fluorescent dye penetrant examinations of the weld overlays showed that the central part of the weldment contained a certain amount of porosity and sound welds are seen on the peripheral areas of the repair weldment. If the depth of the cavitation damage is superimposed on the repaired areas, it is found that the weldments are sound on areas where the depth of cavitation damage is less than 1.5 mm and excess amounts of porosity are seen on repaired areas where the depth of the cavitation damage is more than 1.5 mm. The repaired weldments containing Stellite 21 and Hydroloy HQ 913 filler metals consistently showed higher amounts of porosity than those containing ER308 filler metal. As pointed out earlier, the cored filler metal of Stellite 21 and Hydroloy HQ 913 may be causing this defect in the corresponding weld metal.



a b

Figure 4 Different stages of cavitation repair (a) cavitated surface (b) weldment

Conclusions

(1) Cavitating fluid jet method can be used to simulate the early stages of cavitation erosion(Frosting) on hydraulic turbines.

(2) Results of weld overlay experiments on the cavitated specimens and subsequent field tests suggest that Pulse Gas Metal Arc welding procedures can be applied to overlay directly on the cavitated areas where the depth of damage is less than 1.5 mm. These procedures are applicable to the horizontal or vertical position of welding.

Recommendation

The following optimum weld overlay procedures are recommended to repair cavitation damage on hydraulic turbines without gouging or grinding the cavitated areas. These procedures are only applicable to the cavitated areas where the depth of cavitation damage is less than 1.5 mm.

Base Metal of application: ASTM A 743 grade CA6NM Martensitic Stainless Steel Casting
Austentic Stainless Steel weld overlay (308)
Cobalt based weld overlay (Stellite 21)

Low cobalt containing austenitic stainless steel weld overlay (Hydroloy HQ 913)
Inspection: Examine the cavitated surface for cracks and other casting or weld defects. These defects, if found, shall be removed by grinding and the ground surface should be re-examined.
Preheat: 100°C (Electrical resistance heating pads are preferred but propane torches can also be used).
Welding Process: Pulse Gas Metal Arc Welding Process (P-GMAW)
Polarity: Direct Current Reverse Polarity (DCRP)
Shielding Gas: 98% Argon - 2% Oxygen
Flow Rate: 850 litres per hour
Welding Parameters: For Horizontal Applications refer Table 1(a)
For Vertical Applications refer Table 1(b)
Tie-in (Overlap): 35% minimum
Interpass Temperature: 200°C maximum
Profiling: Grind the weld metal to the original contour of the component.
Final inspection: Visual followed by a visible dye penetrant examination. Cracks, lack of fusion defects and porosities more than 3 mm diameter are not acceptable.

Acknowledgement

This work was funded by the Canadian Electrical Association, Montreal, Canada and B.C. Hydro, Vancouver, Canada.

References

1. "Hydraulic Turbine Cavitation Pitting Damage", CEA R&D Report 000 G136, 1981.
2. "Cavitation Pitting Mitigation in Hydraulic Turbines", EPRI Report, AP-4719. Two volumes, 1986.
3. "Hydraulic Turbine Overlay Repair Process Without Gouging", CEA R&D Report 229 G594, 1992.
4. "Metallic Overlay Materials for the Optimum Cavitation Performance of Hydraulic Turbines", CEA R&D Report 135 G273, 1987.

REPAIR OF CRACKING
IN
HIGH HEAD (2,378') PENSTOCK

Patrick J. Regan[1]

ABSTRACT

Balch 1 powerhouse is a 34 MW hydroelectric facility located on the Kings River approximately 40 air miles east of Fresno, California. The plant was constructed in 1926 and placed in service in early 1927. The turbine generator is a double overhung, horizontal shaft, impulse unit. The gross head on the plant is 2,378 feet and the normal maximum flow is 213 cfs.

The water delivery conduit includes 3.7 miles of 12 foot tunnel and 4,882 feet of steel penstock. The penstock varies from 60 inch diameter at the top to two 34 inch pipes at the powerhouse. The upper 2,596 feet of penstock is plain forge welded steel. The next 2,286 feet consists of banded forge welded pipe. The elbows and other special fabrications such as manholes and expansion joints are forge welded fabrications in the upper reaches of the penstock and castings in the lower portion. The castings have integrally cast flanges for connection to adjacent sections.

The penstock is supported on concrete saddles in the upper reaches. The bottom 300 feet of twin 34 inch pipe is fully encased in concrete anchor blocks. The 34 inch pipes cross under the Kings river to deliver water to Balch Powerhouse.

On November 26, 1990 powerhouse crews noticed water leaking from the anchor block adjacent to the powerhouse. Circumferential cracking was found on the inside surface of the cast sections near the flanges. Non-destructive testing indicated that some of the cracking was through wall.

---

[1]Senior Civil Engineer, Pacific Gas & Electric Company, San Francisco, California

Based upon the non-destructive examination, fracture mechanics analysis, and engineering evaluation, PG&E removed and replaced the castings below the wye branch (with the exception of the manhole sections on the north side of the river) with new fabricated sections.

Construction problems included difficult access, steep terrain, working within the Kings River, a short construction schedule and temperature extremes. The project was engineered and constructed by PG&E forces with the exception of pipe erection which was performed by a contractor.

BACKGROUND

Balch 1 powerhouse is a 34 MW hydroelectric facility located on the Kings river approximately 40 air miles east of Fresno, California. The plant was constructed in 1926 and placed in service in early 1927. The turbine generator is a double overhung horizontal shaft impulse unit. The gross head on the plant is 2,379 feet and the normal maximum flow is 213 cfs.

The water delivery conduit includes 3.7 miles of 12 foot tunnel and 4,882 feet of steel penstock. The penstock varies from 60 inch diameter at the top to two 34 inch pipes at the powerhouse. The upper 2,596 feet of penstock is plain forge welded steel. The next 2,286 feet consists of banded forge welded pipe. The elbows and other special fabrications such as manholes and expansion joints are forge welded fabrications in the upper reaches of the penstock and castings in the lower portion. The castings have integrally cast flanges for connection to adjacent sections.

The penstock is supported on concrete saddles in the upper reaches. The 300 feet of twin 34 inch pipe is fully encased in concrete anchor blocks. The 34 inch pipes cross under the Kings river to deliver water to Balch Powerhouse.

On November 26, 1990 powerhouse crews noticed water leaking from the anchor block adjacent to the powerhouse. The jet of water coming from the anchor lock was approximately 6 feet high. The powerhouse operators closed the penstock guard valve, dewatered the penstock and opened it for inspection. PG&E engineers and metallurgists inspected the pipe and uncovered no obvious metallurgical defects. An O-ring seal appeared to be deteriorated in an area adjacent to where the leakage was occurring. This seal was removed and replaced with a new neoprene O-ring. When the pipe was subsequently refilled, the leakage returned in the same area and quantity.

On December 10 an effort was made to locate the leak location by acoustic emission testing. The testing was relatively unsuccessful. A slight leak was detected in the "A" (easterly) penstock between the TSV and the manhole. No other leaks were detected.

The failure of the acoustic emission testing to locate a single major leak source was interpreted as an indication that several minor leak sites were present. A decision was therefore made to line the penstock with a polyurethane coating. While sandblasting to prepare the penstock for lining a series of hairline cracks were discovered in the cast steel sections of penstock.

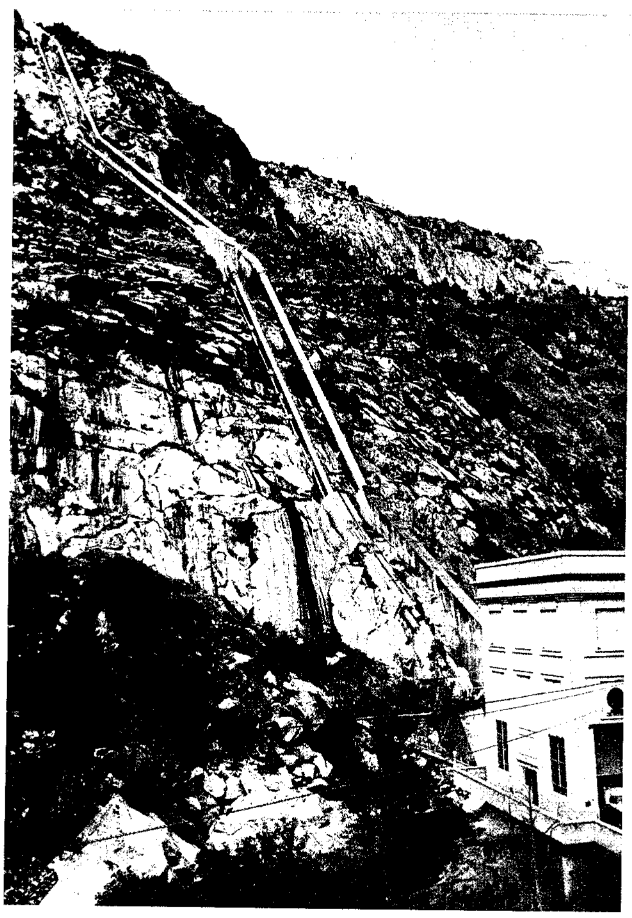


Figure 1 - Balch 1 Powerhouse
Balch 1 penstock is the pipe on the left side of the picture. Repairs to cast steel sections were made in the concrete encased section located at the bottom of the penstock.

CASTINGS

The special sections in the lower section of the penstock were fabricated from cast steel. The various sections are joined by bolted flanged connections. The flange detail incorporates two O-rings to seal the joint. The design allowed one of the O-rings to be removed and replaced from inside the penstock if necessary.

Cast steel tends to be more brittle than rolled steel and is susceptible to a variety of flaws including porosity and shrinkage cracking. Shrinkage cracks occur most frequently in areas where there is a significant change in cross sectional area. In the penstock castings the flanges are cast integrally with the pipe shell. The flanges are 7-1/2 inches thick while the pipe wall is roughly 2-3/4 inches. The thickness of these sections allows the exterior of the casting to cool while the interior is still hot. Subsequent cooling of the casting core causes thermal stresses in the outer layers of the casting. These stresses can cause cracking in the castings.

The 65 years of service has subjected the shrinkage cracks in the penstock to corrosion. The corrosion has acted to enlarge the original shrinkage cracks in both length and depth. The installation procedure for the pipe may also have contributed to increased cracking. Installation of the pipe during hot weather followed by the subsequent introduction of cold water would cause the pipelines to shrink. The concrete encasement prevents the pipes from shrinking fully and thereby causes longitudinal tension stresses. The longitudinal stresses would act to enlarge the circumferential cracks.

INSPECTION PROGRAM

Following the discovery of these cracks PG&E abandoned the lining operation and undertook an extensive inspection program. All of the cast steel sections and forge welded elbows were sandblasted and inspected by magnetic particle and ultrasonic examination. The majority of the castings had circumferential cracks located adjacent to the flanges. A longitudinal crack was found in an expansion joint.

Magnetic particle testing (MT) consists of applying a magnetic field to the pipe wall through a MT yoke. The legs of the yoke are placed on each side of a suspected crack. The magnetic current generates a magnetic field in the pipe wall. The crack aligns the magnetic field such that when a magnetic powder is sprinkled on the steel the powder is held in the crack.

Ultra-sonic testing (UT) uses sound waves to measure pipe wall thickness and crack length and depth. A sound wave is generated and applied to the pipe wall by a transducer. The sound waves bounce off discontinuities such as the pipe wall or a crack. The time it takes for the sound to return to the transducer is measured. The return time is then translated into a distance to the discontinuity. To measure wall

thickness straight beam UT is used. This places the transducer at right angles to the pipe wall and the sound goes straight across the pipe wall. Crack depth is measured using angle beam UT. In this technique the transducer places the waves into the pipe wall at an angle so that they will intercept the crack. The transducer is moved back and forth along a line perpendicular to the crack. This allows the operator to determine where the crack tip is located. At depths below the tip, the waves are not reflected to the transducer.

FRACTURE MECHANICS

In parallel with the penstock inspection program, PG&E contracted with Structural Integrity associates to perform a fracture mechanics analysis of the penstock. This analysis provided a critical flaw size against which the actual flaws could be judged.

The fracture mechanics analysis included a finite element model of the lower portion of the penstock to determine applied stresses, mechanical testing to determine physical properties of the cast material, and a computer analysis of crack mechanics.

The finite element program modeled the lower portion of the penstock from the bend U/S of the manholes on the far side of the river to the TSV. It included loads applied due to the pressurization of the penstock as well as loads that were induced by temperature changes between the installation and operation of the penstock.

A sample was removed from the "A" penstock bend reducer for mechanical testing. The testing included Charpy V-Notch impact tests, tensile tests, and a chemical analysis. Results of the testing are presented in Table 1.

The Charpy V-notch values are extremely low compared to materials that provide good fracture toughness. They indicate that the cast steel in the castings is susceptible to brittle fracture and that the critical flaw size will be fairly small. The yield and tensile strengths are reasonable for cast steel of this era. The elongation however, is very low and shows that the material has very little ductility prior to failure.

The chemical analysis shows relatively high carbon, phosphorus and sulfur indicating that the cast steel is not very weldable. This resulted in a decision to preserve the flange connections on the pipe sections that remained in place and fabricate flanges on the new connecting pieces.

The result of the fracture mechanics analysis shows that the critical flaw size is about 2 inch long for a through wall crack and about 1/4 inch deep for a full circumferential crack. The critical flaw sizes include a 2:1 factor of safety.

Charpy V-Notch	7.4 foot pounds @ 32oF; ave of 27 tests 5.0 foot pounds @ 32°F; low of 27 tests	
Yield Strength	37,150 psi	
Tensile Strength	65,700 psi	
Elongation	15.2%	
Chemical Analysis		
	Carbon	0.23%
	Manganese	0.47%
	Silicon	0.29%
	Phosphorus	0.010%
	Sulfur	0.008%
	Chromium	0.01%
	Nickel	0.02%
	Molybdenum	0.01%
	Vanadium	0.001%
	Copper	0.05%
	Iron	Remainder

Table 1 - Results of Lab Testing of Balch Penstock Material

RECOMMENDED REPAIRS

The cracked sections in the "A" (easterly) and "B" (westerly) penstocks that are encased in concrete were replaced with new fabricated steel sections prior to returning the plant to service. The recommendation to replace these sections prior to returning the plant to service was based upon the high pressure in this section, the thick casting sections which are more likely to have severe shrinkage cracking, the fact that leakage would be difficult to detect because of the concrete encasement, and the danger to life and property if one of these sections were to fail.

Both the "A" and "B" penstocks from the TSV's through the elbows on the powerhouse side of the river crossing were removed and replaced with new fabricated steel sections. This section was most likely to endanger life and property at Balch if a failure was to occur.

Bend-reducer sections are located immediately upstream of the TSV's and transfer the thrust of the closed TSV into the concrete anchor block 16. The TSV itself is not anchored independently to the powerhouse and the original cast steel bend-reducers did not have thrust rings other than the flange to transfer the thrust loads.

If the bend reducers were to fully fracture while the TSV was closed, it was possible for the TSV to pull out from the anchor block. This would release a high pressure jet of water into the powerhouse causing extensive damage to the powerhouse, and possibly loss of life, before the penstock guard valve could close.

The elbow sections located upstream of the bend reducers on both the "A" and "B" penstocks also had cracking of various degree. Although it is unlikely that a fracture in this area would have resulted in the penstock pulling out of the concrete block, if the cracks grew to be through wall, high pressure water pockets could develop in the concrete. The concrete anchor is lightly reinforced, of poor quality and susceptible to failure if a high pressure water pocket developed over a large enough area. The failure would most likely involve blowing out chunks of concrete from the anchor and a possible subsequent failure of the penstock. The failure of a pipe would result in a high pressure water jet and the release of large quantities of water.

The elbows on the powerhouse side of the river crossing in both the "A" and "B" penstocks were severely cracked. In some cases the cracking was through wall. These elbows needed to be replaced in order to maintain the integrity of the penstock system.

In order to replace the cast sections from the manholes to the TSV and the cast elbows at the river crossing, it was necessary to excavate the majority of anchor block 16. The excavation uncovered portions of the straight sections between the manholes and the lower elbows. If these sections were to be reused, hand excavation would have been required near each end in order to preserve the flanges at each end of the straight sections. In addition the removal and replacement of the lower elbows would be more difficult since there would be limited room to maneuver the new sections into place.

PG&E decided to remove the straight sections. This allowed a quicker project schedule, reduced the amount of hand excavation, and was estimated to reduce the total direct cost of the installation. The excavation was quicker and cheaper since a hoe ram was used to excavate the concrete with only minimal hand work. Installation of the new pipe was also simplified since the work area was not restricted. The replacement of the straight sections also completely renewed the penstocks on the powerhouse side of the river crossing.

The elbows on the upstream side of the river crossing had some cracking in the "B" branch. Comparison of the cracks in the "B" penstock to the critical flaw size from the fracture mechanics analysis indicated that these elbows should be replaced. Due to the close proximity of the two pipes and the difficulty of removing the concrete encasement a decision was made to replace the elbows in both the "A" and "B" branches.

The elbows and venturi sections located upstream from the manholes on the upstream side of the river were severely cracked and required replacement. Due to the location of these pieces and the limited extent of the recommended replacement, these sections were replaced through limited excavations.

The wye branch and the elbow section, #14, located upstream of the wye were judged to be safe and did not require replacement.

Inspection of the straight section beneath the river showed one O-ring had been pushed out of its pocket. This was removed and replaced to renew the integrity of the original gasket design.

Some of the castings which were not recommended for replacement have spongy looking areas that are original casting defects. Ultrasonic examination and fracture mechanics analysis showed that these defects did not pose a risk to the penstock. These areas were repaired with an epoxy material to stop future corrosion and erosion from enlarging the voids.

Elbows 5 through 13 are castings which are encased in individual anchor blocks. Elbows 10 through 13 are located on the side of the extremely steep, as much as 60 degrees, north slope of the Kings River canyon. The fracture mechanics analysis showed that the lower head in this section resulted in a larger critical flaw size than in the lower penstock sections. In addition the thinner castings had less severe cracking. Comparison of the critical flaw size with measured flaws indicates that these castings do not require replacement at this time.

An expansion joint located in the upper reaches of the penstock between anchor blocks 3 and 4 had a longitudinal crack that was probably associated with the forge welding fabrication. This joint was replaced with a new fabricated expansion joint in 1991.

An earlier repair was made to the pipe in the vicinity of anchor block 4. An air valve attachment had failed leaving a hole in the penstock. The repair consisted of welding a steel ring circumferentially around the pipe. The inside of the pipe is now experiencing circumferential corrosion on the line of the welds. This repair will be cut out of the penstock and a new can section welded in place during a scheduled plant outage in 1993.

REPAIR PROCEDURE

In order to replace the penstock sections between the river crossing and the TSV's the concrete encasement around the pipes was removed. The removal was done from a work area on the concrete overpour weir. The existing road from Powerhouse 2 to the river was improved to allow equipment access to the weir. The water quality was

protected by importing clean gravel to use as a road base and diverting the stream flows through CMP culverts.

A hoe ram was used to break out the concrete encasement. Careful hand excavation was required around the flange connecting the lowest cast elbow to the upstream straight can. Hand excavation was also used where the bend reducer passes through the powerhouse wall. A structural steel column is supported on a reinforced concrete column located between the "A" and "B" penstocks. The column supports the overhead crane rail and the powerhouse wall. To avoid damage to the column the wall was saw cut to prevent cracking. Similar techniques were employed to prevent damage to the parapet walls around the generator air ducts. The concrete subgrade was finished in a notched shape to provide a suitable key for the new concrete placement.

The concrete surrounding the elbows and venturis on the far side of the river were excavated by hand in order to not injure the adjacent straight sections which will remain. Expansive grout was used to crack the concrete allowing easier removal of the anchor block. Some over break occurred as the grout followed concrete lift lines and the gap between the pipe and concrete.

The replacement steel penstock sections were shop fabricated. The replacement pipe on the powerhouse side of the river was supplied in six pieces. The first section was a flange with an attached short section of pipe that attached to the straight pipe section in the river crossing. The second section replaced the lower two elbows. The third section replaced the straight run of pipe up to the manhole. The fourth section was the upper vertically curved elbows. The fifth piece replaced the horizontal bend reducer. The sixth piece was a flange with attached short section of pipe that will bolt to the TSV and weld up to the bend reducer. The replacement pieces for the elbows and venturis on the far side of the river were individual fabrications in two pieces that allowed for insertion into the excavated openings.

Due to the thick pipe section required, 1 1/2 inch, the difficulty of making the final closure weld, and the need to stress relieve and radiograph the field closure welds, the replacement pipe was supplied by a contractor who also installed the pipe.

After the penstock sections were replaced, a new concrete encasement was placed. The concrete was reinforced and doweled into the existing concrete subgrade.

WORK SCHEDULE

The design for the replacement pipe sections began in March and was completed in May 1991. Fabrication and erection specifications were also prepared and the job was bid in May. The design efforts proceeded concurrently with obtaining management authorization to proceed with the unbudgeted work. Management

approval was given in July and the contract for fabrication and erection was let immediately.

While the penstock sections were being fabricated, PG&E construction crews mobilized to remove the concrete anchor blocks and the damaged pipe sections. The steel fabrication schedule anticipated that the initial replacement sections would be available in six weeks. In order to accommodate the erection of the new pipes all excavation would need to be completed prior to the arrival of the new steel. PG&E crews worked two 10 hour shifts to complete all demolition and site preparation work prior to September..

Fabrication of the steel was completed roughly as scheduled and sections of new pipe began arriving on site in mid September. Installation of the new pipe sections took approximately three weeks including installation, welding, stress relieving and radiography. The pipe on the powerhouse side of the river was installed first to allow PG&E crews to begin forming and placing the concrete anchor blocks and close in the powerhouse prior to the winter rain season. By beginning to place the anchor blocks on the powerhouse side of the river while the contractor completed pipe erection on the opposite side, PG&E was able to complete placement of the anchor blocks by early November.

Following startup testing the powerhouse was returned to service in early December 1991.

CONCLUSION

Many early high head hydroelectric plants have cast steel elbows, manholes and other special sections. The casting process can leave defects that were either not noted or were deemed acceptable at the time of fabrication and installation, and in general provides a material with poor fracture toughness. Installation and use may induce stress that acts on the pre-existing defects in a manner that will open the defects. Corrosion will then work on the cracks acting to enlarge the defect. In some instances the defects may become through wall cracks leading to leakage from the penstock.

The cracks discovered by PG&E in the Balch penstock were all circumferential with the sole exception of the crack in the expansion joint. The circumferential orientation of the cracks was most favorable from a catastrophic failure perspective. The penstock stresses would not tend to separate the penstock sections even if the cracks propagated fully around the pipe. A failure would most probably result in an extreme leak from the pipe but would not necessarily result in a catastrophic failure.

PG&E is undertaking an inspection program to ensure that similar problems do not exist at other facilities.

"TURN KEY" CONSTRUCTION TO OPERATIONS AND MAINTENANCE A TRANSITIONAL PROGRAM

BY

Bob Barksdale   Mike Lewis   Russ Pytko

ABSTRACT

The transition from a construction "Turn Key" environment to a fully operational power plant, with minimum disruptions to the contractor and the owner, requires detailed planning and cooperation from each of the interested parties. Experience on several hydroelectric and cogeneration power facilities ranging from 14 to 192 Megawatts has shown that this transition can be done efficiently and economically by an Independent Power Producer (IPP) with limited resources. Early planning based on firm objectives and with due consideration for concerns of financial entities, dedicated commitments by the IPP management and the development of a comprehensive O&M Plan are key elements in the success of such an undertaking. This article gives an overview of the necessary ingredients for a successful transitional program.

BACKGROUND

Typical "Turn Key" construction contracts for hydroelectric and cogeneration power projects contain clauses with incentives and bonuses for early completion. These could range from set fees on a per day basis to a share of the revenues for power generation over a specified period. The contractor; therefore, strives for early completion with minimum disruptions during start-up, acceptance and turnover of the project in order to benefit from these incentives.

On the other hand, the owner has similar interests with minimizing disruptions in order to meet its commitments to lenders, power purchasers, government and civic entities, etc., by the implementation of an orderly and phased transition from construction to a fully operational power producing facility.

Bob Barksdale is a civil engineer and president of KEYCOR, INC. a professional management consulting firm to the engineering and construction industry.

Mike Lewis is a civil engineer employed by the Bechtel Group specializing in the construction, operations, and maintenance of hydroelectric power projects.

Russ Pytko is an associate with KEYCOR, INC. specializing in contracts administration and project management.

Ideally, conditions for owner acceptance of a "Turn Key" project to insure cooperation and mutual objectives of both the owner and contractor, should be clearly defined in the construction contract. In most cases they are not, thereby creating a dilemma. Quite frequently, construction contracts are also lacking a precise definition of what constitutes project completion. Furthermore, project completion may have different connotations by various entities, e.g., FERC, Utilities, lenders, etc. which may not be congruous with the construction contract. Such inconsistencies add to the potential confusion during the transition period.

TRANSITIONAL PLANS

Separate and concurrent planning for the transition by the contractor and the owner should begin early in the construction phase. The contractor needs to make the necessary plans for resources during the transition and integrate these resources into the preoperational testing, start-up, turnover, acceptance and warranty schedules. Since these activities normally trigger or influence other events, such as:

- Bonus for early project completion
- Revenue bonuses
- Utility acceptance
- Start of equipment, unit, and plant warranties
- Start of performance guarantees
- Release of retention
- Release of bonding, etc.

it is important that all parties are working toward common goals during this period.

The owner also must be prepared to implement an O&M program without interferences and disruptions immediately upon turnover of individual units and eventually, the complete facility. Both parties should periodically review and coordinate their plans since both have a common objective - a successful power producing facility. Due to the expected longevity of the facility and the importance of achieving optimum power production, the development of a viable O&M program by the owner is essential for a smooth and orderly transition. The consequences of the owner not being prepared could be devastating from financial, legal, and professional considerations.

O&M PROGRAM

In establishing guidelines for the O&M program, the owner should, as a minimum, consider various factors, such as:

- Primary and secondary IPP objectives to be accomplished with the program
- Financial entity's concern for O&M requirements
- Limited resources of the IPP
- Requirements, objectives and special interests of both parties during the transition from "Turn Key" to O&M
- Provisions for contract closeout, equipment, unit and plant warranties and performance guarantees
- Schedule for program implementation

Dependent upon the size of the facility and the IPP's staff, the IPP could either select a cadre of management personnel, the nucleus for the eventual O&M organization , or use its internal staff to develop the detailed O&M plan.

A typical plan, based on the above factors would normally provide for the following but may be altered as necessary to fit a specific need:

- Integrated O&M organization to permit economies in staffing and increased productivity
- Selection of management and supervisory personnel
- Employee training and development to include "cross training" of operators and technicians to enhance promotion opportunities, increase the number of back-up operators and to promote a better understanding of the plant equipment and systems
- Formation of administrative, safety, emergency action, operations and maintenance procedures
- Establishment of reporting control systems and report formats for FERC, COE, utilities, lenders, environmental and others
- Development of spare parts inventory with proper identification, location, quantities, unit cost, procurement lead times and maintenance control systems (manual or computer) utilizing a "Just-in-Time" approach
- Structure of O&M, capital and administrative budgets and code of accounts (see Figure 1)
- Provisions for annual outage planning and scheduling
- Methods for repairing and replacing defective and warranty items and warranty protection
- Use of competitive subcontractors to benefit from economies of scale and to minimize overhead and equipment expenses
- Schedule for safety and insurance inspections
- Fostering harmonious relationships with government agencies, utilities and the local community

Although this brief checklist may seem to contain items common to the O&M industry, it does identify essential areas for decision making by an IPP with a determination to develop and implement an O&M program with its own resources. Based on a case study for a large 190 MW hydroelectric power project, the IPP introduced components into the O&M plan which were out of the ordinary and instrumental in the successful transition for "Turn Key" to a fully operational facility.

"SHADOW CREW"

During the start-up and testing phase and for a 3 to 6 month period following the turnover of the plant by the contractor, the plan provided for the IPP to employ the services of at least 4 additional senior operators commonly referred to as a "Shadow Crew". These operators, each with 20-30 years experience in start-up, turnover and operations of large multi-unit hydroelectric power plants on the Columbia and Ohio Rivers, worked with the contractor during the transition as well as participated and oversaw the training of the permanent plant operators.

In addition to the "Shadow Crew", 16 of the O&M personnel were seconded to the contractor during the transition period. This program under the supervision and guidance of the contractor, enhanced the technical training for operators and maintenance personnel as well as reduced the contractor's costs by not having to recruit and train such personnel.

CONVENTIONAL HYDROELECTRIC PLANT
CODE OF ACCOUNTS

This generic Hydro Operations Cost Code System has been established to track operations and maintenance budgets and expenditures. The system uses a three letter alpha prefix and a three number suffix.

The alpha code uses four prefixes for four categories of expenditures:

OPX - Operations
MXX - Maintenance
GAX - General and Administrative
CPX - Capital Projects

The maintenance codes can be further delineated by the two letter plant system codes as below:

MBS.XXX - Battery System
MCA.XXX - Compressed Air
MCM.XXX - Control Monitoring
MCP.XXX - Cathodic Protection
MDG.XXX - Diesel Generator
MDW.XXX - Dewatering System
MCO.XXX - Paging and Communications
MCW.XXX - Cooling Water
MDS.XXX - Drainage System and Pumps
MEL.XXX - Elevator
MFD.XXX - Fire Detection and Protection
MPT.XXX - Plant Trucks and Mobile Equipment
MGA.XXX - Governor Air Compressor
MGO.XXX - Governors
MHF.XXX - HVAC Systems
MHV.XXX - High Voltage Substation System
MLO.XXX - Lube Oil System
MLT.XXX - Lighting System
MLV.XXX - Low Voltage Electrical Dist. System
MMV.XXX - Medium Voltage Elec. Dist. System]
MOH.XXX - Oil Storage and Handling
MHT.XXX - Tools - Hand and Small
MST.XXX - Special Tools for Plant Equipment
MSL.XXX - Stoplogs
MWG.XXX - Wheelgates

The plant system codes match the plant equipment files, the plant inventory system codes and the design drawing equipment I. D. system. This allows maintenance items to be coded and tracked to individual equipment systems.

FIGURE 1

TECHNICAL TRAINING FOR OPERATORS AND TECHNICIANS

In addition to the technical training provided by various equipment suppliers, the soon to be operators and technicians, having gone through a rigorous screening (drug and aptitude testing) and selection process, received off-site training at a similar operational hydroelectric power plant under custody of the IPP. At this point, the candidates participated in classroom work in mathematics, physics, dynamics, hydraulics, etc., hands-on training and operational testing, routine scheduled maintenance assignments and the planning for the annual outage. They also acquired familiarity with as-built drawings, tracing of systems, and an understanding of logic and relationships during operations.

As part of the transition, the candidates were assigned for a 6 month period to a quality assurance group assisting the owner's engineer in performing inspections of the project work. Thereby, increasing their knowledge and familiarity with equipment and plant systems as well as contributing to the quality of the end product. Throughout the training period, emphasis was placed on imparting to the trainee an awareness of pride in his work, plant cleanliness, plant performance and the IPP organization.

ANNUAL OUTAGE PLANNING AND SCHEDULING

An O&M program would not be complete without provisions for annual outages, albeit such provisions are not necessarily essential for implementation during a typical transition period with the important exception of warranty protection. They need to be considered; however, in the overall program since planning for the first outage should begin as soon as the facility is turned over to the owner, if not earlier. Based on another case study for a smaller plant under custody of the IPP, several principles were developed and implemented during successful annual outage undertakings; such as:

- Initiation of early, continuous and detailed planning based on maintenance requirements, equipment, unit, and plant warranty provisions and personnel availability through the use of management tools, e.g. work breakdown structures, bar charts, precedence diagrams, trend analysis, etc. (SEE FIGURE NO. 2)
- Flexible scheduling of outage during expected minimum flows and by unit to minimize down time and lost revenue
- Development of detailed schedule within tight work windows for each unit to include duration of each work activity, manpower required, and equipment and spare parts requirements using "just in time" planning for spares (SEE FIGURE NO. 3)
- Determination of needs and lead times for technical representatives for special systems and equipment
- Employment of qualified and competitive contractors to augment maintenance personnel and to expedite completion of work activities
- Development of detailed budgets for the outage within constraints of overall O&M budget
- Conduct of periodic "What if" planning critiques within development of necessary contingencies

OUTAGE SUMMARY SCHEDULE

```
SCHEDULE FOR 1990 OUTAGE
Dependencies:        Plant Shut Down
                     90
                     AUG
WBS Task Name        20   21   22   23   24   25   26   27   28   29   30   31

  PREOUTAGE WORK     ============
Infrared Scan        ===
Station Svc #2          ====
Calibrate SS #2 meters      ===
Calibrate OG meter          ===
Check and Test U#2 Relays      ===
Station Svc #1                    ====
Calibrate SS #1 meters                ===
Check and Test U#1 Relays                ===
Clean Stop Log Slots    ====
Annual Insp. and Svc. Diesel Generator ====

  Unit No. 2 Outage            ======================================
   Shut down U-2               =
   Close Draft Tube Gate        =
   Set Stop Logs                ==
A  Dewatering U-2                 ======
B  Turbine Inspection                     ===============
C  Wicket Gate Insp                       ==========
D  Oil System Insp                        =============
E  Generator Insp                         =======
F  Excitation Insp                               =======
G  Governor Insp                                  ===========
   Water Up U-2                                           ==
   Remove Stop Logs                                         ==
H  Unit Start Up                                              ====

                       90
                       SEPT
                        1  2  3  4  5  6  7  8  9  10   11   12   13   14
  Unit No. 1 Outage==============================
   Shut Down U-1       =
   Close DT Gate        =
   Set Stop Logs        ==
A  Dewatering U-1         ======
B  Turbine Inspection          ===============
C  Wicket Gate Insp            ==========
D  Oil System Insp             =============
E  Generator Insp              =======
F  Excitation Insp                    =======
G  Governor Insp                        ===========
H  Water up Unit                                   ====

  PLANT OUTAGE                                       ======
  Plant Shutdown                                     =
  Diesel Generator in service                        =
  Main Transformer Inspection                        ===
  Trip Test Substation Relays                        ===
  Trip Test Unit Protective Relays                   ===
  Dielectric Test Bus Ducts                           ==
  Restore all systems                                   ===
  Return Plant to service                                 =
```

FIGURE 2

SCHEDULE FOR 1990 OUTAGE **WBS-U#2-A UNIT DEWATERING**
Dependencies: Plant Start Up (Resource Schedule)

Task Name	Resources (Manpower)	Durations (hours)	Effort (M.H.)	Notes
Dewatering of Unit #2	Operators	34.0	1.0	
Unload Scaffolding	Maint-1, Maint-2	0.0	0.0	Prior to August 20, 1990
Remove Stoplog Grating	Man-1, Man-2, Man-3	2.0	6.0	
Install Stoplog Handrails	Man-1, Man-2, Man-3	1.0	3.0	
Remove Debris	Maint-1, Man-2	8.0	16.0	
Turn on Air to U/S Gate Guides	Operator	0.1	0.1	
Turn on Air to D/S Gate Guides	Operator	0.1	0.1	
Shut Down Unit #1	Operator	0.5	0.5	
Close D/S Gate	Operator	0.1	0.1	
Clear Unit for Divers	Maint-1, Maint-2	1.0	2.0	Servomotors locked/secured
Divers Insp. Gate Guides and clear Stop Log Slots	Diving S/C-3 man crew	2.0	6.0	
Rig Crane	Man-4, Man-5, Man-6	2.0	6.0	To lift Stoplogs
Divers Assist Setting Stop Logs	Diver Crew - 3 men			
	Maint-1, Maint-2, Man-1	8.0	48.0	
Open Air Valves to U/S Gates	Operator	0.1	0.1	
Install Water Level Tubing	Maint-1, Maint-2, Man-1	1.0	3.0	
Align Valves for Dewatering	Operator	0.5	0.5	
Close Air Valves to U/S Gates	Operator	0.1	0.1	
Crack open and Close D/S Gate	Operator	0.1	0.1	
Begin Dewatering of U#1	Operator	0.1	0.1	Complete during the evening prior to 0800 on August 25
Complete Dewatering	Operator	0.1	0.1	
Tag out Draft Tube Gate	Maint-1	0.5	0.5	
Open Draft Tube Access Door	Maint-1, Man-2	2.0	4.0	Unit Water passage ready for inspection

FIGURE NO. 3

Detailed planning and scheduling for annual outages is an arduous task and staff disciplining process but the rewards can be significant not only in the short term but throughout the life of the facility. Proven techniques and features incorporated into the planning process can significantly influence the results of the outage with the results measured by minimum down time in the future and high equipment, unit, and plant reliability and efficiency.

CONCLUSIONS

Making the transition from a "Turn Key" construction environment to a fully operational power plant can be accomplished effectively, efficiently, and economically by an IPP with limited resources and experience. Detailed planning and continuous cooperation and coordination between the contractor and the owner are necessary for a successful transition. Even though the contractor and the owner have different objectives, they strive for a common satisfaction in a reliable facility capable of producing electrical energy.

From the owner's standpoint, the transition can be facilitated with the development of a viable O&M program based on firm objectives and a determination of the IPP management to accomplish these objectives with their own resources. The O&M plan must be developed early, implemented and refined during the transition period and in sufficient depth and detail to be applicable when the facility is fully operational. Extraordinary components of the plan, i.e., use of a "Shadow Crew", O&M subcontracts, staff disciplining and special technical training for operators and technicians can be useful in a successful transition. The success can be measured by the avoidance of claims, improvement in IPP's preparedness, reduction in costs, increases in unit availability and revenues and assurances of high degree of personnel and equipment safety.

BOTTOM LINE

The IPP must be properly and thoroughly prepared to complete, accept, operate and maintain the facility. Otherwise, the consequences could be devastating, such as:

- Liability for contractor's claims
- Losses of warranty protection
- Losses of insurance protection
- Non-compliance orders from FERC, Utility, Lenders, and others
- Losses of revenues and profits

Talking Trash: Debris Removal at Hydropower Plants

James F. Sadler, Jr. P.E.[1]

Abstract

The accumulation of trash and debris on turbine intakes at hydroelectric powerplants is a universal problem. The Cheatham Powerhouse is a specific example of this problem. Cheatham is a low head, main-stem project with a powerplant designed for regulated flows from upstream reservoir projects. The invert of the power intake structure is at river bed elevation. Water flow passes through the powerhouse with approach velocities sufficient to draw waterlogged trash and floating debris onto the intake trash racks. Consequently, trash accumulates on the racks and floor of the forebay. This clogging reduces efficiency of the turbines resulting in lost revenues from lower power generation capability.

Introduction

In 1963, a floating trash boom was provided in front of the powerhouse. The boom was successful in preventing floating trash from reaching the power intake, but ineffective in catching the waterlogged material that washed under the boom. Floating debris was periodically flushed through spillway gates to reduce safety and operating problems at the project. Debris that was too large or waterlogged was dredged out biannually using floating plant. The debris was loaded on a barge, floated upstream, unloaded and secured on the river bank. The Nashville District of the Corps of Engineers was given the opportunity to implement a test program applying new technology to this hydropower plant.

[1] Project Manager, Engineering Division, United States Army Corps of Engineers, Nashville District, P.O. Box 1070, Nashville, TN 37202-1070, Attn: CEORN-EP-P

Design Criteria

The system was designed to lift a diversified collection of debris ranging from refuse, bottles and other small items to entire trees. Several trash removal systems were evaluated and deemed too limited in their applicability. These included catenary and claw-type removal systems (the debris kept slipping through the lifting mechanism). The system chosen for removing trash from the turbine intake areas used water velocity to move the debris toward the trashrack and then teeth or pins on motor driven chains lifted the trash (both submerged and floating) to the conveyors and subsequently to a collection area. The continuous operation enhanced the system effectiveness. Debris bound on lifting pins would be released from that pin and lifted by a subsequent pin. Furthermore, the racks had to be large enough to cover all three intakes, approximately 60 feet high by 25 feet wide. Conceptual layouts of the system and preliminary computations assured that self-cleaning trashracks could be constructed large enough for this project.

Procurement

Sufficient expertise in the private sector existed to warrant a Request For Proposals (RFP) through the competitive negotiation process. Conceptual drawings and performance specifications were developed and the RFP was issued in January 1990.

Specifications

The RFP required a 50% design submittal with the successful bidder responsible for final design. The trashrack was to lift a 36" diameter, 60 foot long log as well as mats comprised of debris such as vegetation, bottles and tires. The trashrack was required to operate continuously with a reset feature that would allow oversized material to be released from binding. A conveying system which would move the lifted debris to a collection bin was also required. The one rigid requirement for these trashracks was painting. The high degree of rubbing as well as the continuous submergence of the trashracks were our main concerns. The Corps' Construction Engineering Research Laboratories (CERL) recommended an initial coat of zinc-rich vinyl and subsequent coats of vinyl with added abrasives.

Proposals

Initial proposals were submitted on 13 April 1990. The Duperon Corporation of Saginaw, Michigan was issued a Notice To Proceed on 10 December 1990 with a 420 calendar day completion period. The trashracks were completed in February 1992 at a total construction cost of $5,140,000.

Structural Design

The Duperon Corporation subcontracted Spicer Engineering of Saginaw as their structural consultant. They were responsible for the design of the main trusses, horizontal and inclined conveyors, face racks, foundation design, anchorages and concrete trash storage bin. The steel members were designed in accordance with A.I.S.I. standards.

Mechanical Design

The trashrack driveshafts, driveshaft bearings, hydraulic equipment and chains were designed by Duperon Design Incorporated. The shafts were evaluated for loading, stress, deflection and fatigue. The bearings were designed for a 100 year continuous use life. The hydraulic system was designed with over-pressure protection. The chains were designed to individually withstand the entire maximum load based on over-pressure.

Electrical Design

Schwarderer and Associates, the electrical design firm, provided the following:

1. Trashrack operation sequence

2. Electrical layout and conduit list

3. Wiring diagrams for the control system with integrated Programmable Logic Controller (PLC)

4. Control panels layout and detail drawings

5. Computations for motor feeder circuit conductor, motor branch circuit breakers, motor control center main breaker and circuit voltage drop

6. Annunciation and alarm management scheme

7. PLC ladder diagram

Prototype

One of the superior points of the Duperon Corporation submittal was the proposal to build a prototype trashrack prior to fabrication. This allowed the Corps to verify the workability of the Duperon design. The Duperon Corporation was able to substantiate dimensional tolerances for the trashracks from the prototype.

Fabrication

All steel was fabricated in the Saginaw, Michigan area. Prior visits by Nashville District personnel to local machine shops established their capabilities. The initial quantities of steel were estimated to be one million pounds. However, these quantities escalated to 1.4 million pounds by the end of fabrication.

Assembly

After fabrication the material was moved from Saginaw to a local assembly area 16 river miles upstream of the project. The fabricated steel filled 34 tractor trailers. All of the racks as well as the conveyor were assembled here. The hydraulic drives were also mounted at this location as well as the lifting chains. Approximately 25,000 feet (5 miles) of chains were used on this system. One section was operated prior to site delivery. This section successfully lifted two test logs of the specified diameter.

Erection

All of the material was moved from the assembly area to the erection area via barge. Six barges were required to move this material. Erection was facilitated via a barge-mounted crane. Divers set mounts on the inlet forebays and then guided the individual trashrack sections to the mounts. After this operation was completed, the horizontal conveying rack sections were placed. A concrete trash storage bin and the electrical wiring and controls were installed concurrent with those operations.

Acceptance Test

An operational test was required prior to acceptance. The trashrack system had to lift the design log from the river and convey it to the storage bin. The system substantially met the required RFP acceptance criteria in February 1992.

Field Experience

The trashracks have been in operation for over a year. Operating personnel say that the system is removing 95% of the debris impacted on the trashrack. As in any test program, bugs must be worked out of the system. The hydraulic drive system had an initial overheating problem which were resolved by increasing the diameter of the pipe. The chains would jump out of the drive gears when impacting bound material. Metal guides were installed to alleviate this problem. Some debris was dropping between the slots of the rack as the trash is being lifted from the river. This issue is still being

addressed.

Summary

The Cheatham Self Cleaning Trashrack system is now fully operational. The installed system meets the intent established in the specifications. The Competitive Negotiation process decreased the contract time in a cost effective manner.

ESTIMATING QUANTITIES OF ORGANIC DEBRIS ENTERING RESERVOIRS IN BRITISH COLUMBIA, CANADA

Ken Rood, Niels Nielsen and Brian Hughes[1]

Abstract

For their Dam Safety program, B.C. Hydro is preparing debris management plans for their reservoirs in British Columbia. These include an inventory of debris currently on the reservoir as well as an assessment of potential inflow volumes from the catchment, particularly during large storms.

Few measurements have been taken of debris inflow quantities to reservoirs and the program assessment relies on estimating the supply from different sources, such as tributary streams, landslides on reservoir slopes and blowdown and erosion along the shoreline.

Introduction

The B.C. Hydro system consists of 60 dams at 43 locations in British Columbia and their storage reservoirs range from small run-of-the-river headponds to Williston Lake, which at 1,780 km^2 is the largest body of fresh water in B.C. (Figure 1). Debris on these reservoir poses a number of problems. It affects recreational use by powerboats and float planes and may restrict commercial navigation. Accumulations of debris pose a fire hazard, may accelerate erosion of private and public property along the reservoir shoreline, or may damage or restrict access to habitat.

Accumulation of fine debris on power intake trashracks lowers efficiency resulting in reduced power production and increased costs for maintenance. However, dam safety is the main concern and blockage, or damage to,

[1] Ken Rood and Brian Hughes, Northwest Hydraulic Consultants Ltd. #2-40 Gostick Place, North Vancouver, B.C. Canada. V7M 3G2. Niels Nielsen, B.C. Hydro, 6911 Southpoint Drive, Burnaby, B.C. V3N 4X8.

discharge facilities and overtopping and potential failure of the dam is the greatest potential hazard associated with reservoir debris. (Nielsen 1992).

Figure 1: Regions in British Columbia

Debris Management Programs

B.C. Hydro requires their production offices to prepare debris management programs. These programs start with an inventory of debris then proceed with a review of debris issues, an assessment of the risk of dam failure from debris, and finish with preparation of the debris management plan.

There are two components to the inventory: measurement of quantities on the reservoir and assessment of the quantities that may enter the reservoir. The volume, typical lengths and sizes of debris on the reservoir can be estimated during an helicopter overflight or measured from low level air photographs. However, debris inflow cannot be measured directly. Instead it is estimated by the procedures described in the following sections.

Quantifying Debris Inflows

Organic is divided into large organic debris (which includes all stems and branches over 10 cm in diameter) and small branches, twigs, bark, needles, leaves and fine organic mulch which constitute small organic debris.

The organic debris is further classified into fixed (or rooted) debris, found in reservoirs that were not completely cleared prior to flooding, and floating

debris. The volume of floating debris on a reservoir is reduced by removal and disposal programs by B.C. Hydro or the Ministry of Forests, and by passage though the spillway, or other discharge structures and out of the reservoir. The volume also is also reduced by decomposition (a very slow process in freshwater reservoirs), physical abrasion and waterlogging.

Inflow debris arrives from shoreline processes, primarily erosion and blowdown and re-floating of debris, tributary streams and landslides. When calculating debris inflow, the reservoir catchment is divided into reservoir slopes -- that part of the catchment leading directly to the reservoir that provide landslide debris without intermediate fluvial transport -- and tributary drainages, where debris is transported to the reservoir by streams.

Regional Variation in Organic Debris Supply

The supply of organic debris to reservoirs in the B.C. Hydro system varies with climate, geology, physiography and forest cover. The relative importance of different supply processes to the total supply of organic debris, as well as the type and quantity of organic debris material, varies greatly over the Province.

Figure 1 shows the regions used in estimating organic debris supply which are generalized physiographic zones. The regions partly incorporate biogeoclimatic zones which describe forest types, tree sizes and wood volumes on slopes. The biogeoclimatic zones (Ministry of Forests 1992) follow elevation bands and most drainage basins contain several zones that grade into each other with increasing elevation.

Debris Supply from the Shoreline

Shoreline regression results from landsliding related to reduced stresses along the flooded toe of slope or development of beaches in erodible material. Shoreline erosion and regression of steep slopes to develop beaches, is strongly affected by materials. In British Columbia limited erosion occurs along shorelines composed of cobbles and boulders, and very rapid beach development and shoreline regression occurs in fine sands and silts. Rates exceeding 500 m in 15 years were observed in fine sands along Williston Lake after impoundment (Morgan 1982). The contribution of debris to the reservoir can be calculated from typical forest cover volumes and estimates of annual shoreline retreat rates. Average retreat rates may be obtained from B.C. Hydro studies, where erosion has been persistent or caused damage, or from historic air photos.

In reservoirs, which are aligned with dominant storm directions, blowdown may be large, particularly during major storms. Trees within one-half of a

typical tree height may potentially enter the reservoir and the proportion of trees susceptible to blowdown during a major storm, or annually, may be estimated from examination of the forest along the reservoir shoreline. The total volume entering the reservoir is calculated from stand densities along the shoreline, a forest equal to one-half of a typical tree height, the length of reservoir shoreline, and the estimated proportion that may blowdown.

Debris Supply from Tributary Streams

Large organic debris supply from streams to reservoirs depends on the rate of input to the streams, storage in the streams and transport rates under the prevailing hydrologic regime. Streams are classified as small (tree lengths are up to 4 to 5 times channel width), medium (tree lengths are roughly equal to channel width) and large (tree lengths are much smaller than channel width), and this classification has implications for debris transport.

Supply of Organic Debris to Streams

Living or dead trees enter streams as a result of knockdown by wind, bank erosion, soil creep and rapid mass movements. Small streams often have narrow, or non-existent, valley flats and receive debris from windthrow on the lower valley walls and directly from debris torrents and slides.

Medium and large streams, with a well-developed valley flat, receive their organic debris load primarily from bank erosion and tributaries and, to a much lesser extent, from windthrow and mass movement on valley walls. Supply from bank erosion is highly variable along the channel and is driven, in part, by the exchange processes associated with transport of coarse bedload sediment. Annual bank erosion rates can range from zero in bedrock-bound channels, to five ha per kilometre of channel in large, anastomosed rivers, such as the Skeena River (Beaudry et al 1990).

The channel pattern of a river provides a rough indication of bank erosion rates. Lateral activity in large, alluvial rivers is greatest in unstable, braided pro-glacial rivers and least in bedrock-bound or canyonized channels. Between these extremes, lateral activity increases for rivers with straight to irregular patterns, is greater in meandering channels and higher still in wandering and anastomosed channels. Generally, the supply of debris from bank erosion on large streams is best calculated from bank displacements apparent on maps prepared from historic aerial photographs.

Storage of Large Organic Debris in Streams

Stored volumes reflect the capability of the stream to mobilize the supplied debris and also the type and age of the surrounding forest and vary widely from stream to stream. Along the Oregon Coast, Harmon et al (1986) quotes volumes ranging from 7 to 1400 m^3/ha, while Hogan (1986)

measured up to 800 m^3/ha in the Queen Charlotte Islands. Streams in logged areas, or streams that have been cleared or salvaged-logged, have much lower volumes per unit channel area and, typically, smaller, more mobile pieces. Storage quantities along streams can be best estimated during a helicopter overflight.

Transport of Large Organic Debris by Streams

There are few measurements of transport rates of large organic debris. Most available measurements are based on trapping of debris in lakes and reservoirs or on quantities of debris removed from streams following floods.

In small streams many debris pieces are stable and only small pieces are moved by the stream. Stream depth, which is often less than log diameter also limits potential transport rates. In these circumstances decay and abrasion are the most important processes that reduce storage quantities.

In medium sized streams, where channel width is roughly equivalent to the length of typical organic debris pieces, transport is more frequent but distances of transport may be short essentially from jam to jam along the stream. In large streams, depths are sufficient to float material and organic debris stored on bars or in the channel moves annually, or more frequently.

Distance of transport may still be limited as large organic debris, particularly pieces with rootwads, ground on bar surfaces and riffles. Even in large streams the reach contributing debris during any particular storm may only extend a few to a few tens of kilometres upstream from the reservoir.

The distance of travel is partly related to the duration of the high water and travel distances will be much shorter at the annual flood than during maximum floods. Special conditions may increase or decrease typical values. Canyons, lakes or debris jams may trap debris and limit the length of the reach providing debris to the reservoir.

Estimation of Volume Supplied to Reservoirs

Given the lack of empirical data on transport rates, we estimated large organic debris transport from storage along the channel, contributions to the channel from bank erosion and a typical step length, or transport distance,during movement. This approach is very similar to that used to estimate bedload transport rates on rivers from bank erosion (Neill 1971).

The major difficulty lies in specifying the typical step length, or travel distance. On medium-sized streams the typical step length is small, probably only ten or twenty channel widths. On larger streams flow depths during floods exceed typical diameters and loose organic debris stored on bars may travel considerable distances, up to 10 or 15 km or more, as

observed on the Cheakamus River during a large flood and step lengths of 25 to 50 widths should be assumed for large streams. We expect that these typical values will be adjusted after experience in calculating transport rates and comparing these to debris accumulations in reservoirs.

Bank erosion may be a very important source of organic debris during large storms on some streams (Knighton 1987). In very active streams, the erosion may amount to a considerable percentage of the channel width, though there are few observations in British Columbia.

Debris Supply from Reservoir Slopes

Mass movements or slope failures on the reservoir slopes erode and transport forest cover directly to the reservoir. Varnes (1978) classifies mass movements primarily by the type of movement (falls, topples, slides, spread and flows) and secondarily by material type (bedrock, debris or earth). In mountainous areas of British Columbia soils are shallow and consist of coarse material and debris slides and debris flows (called "debris torrents" when confined to gullies and transporting coarse material and organic debris) are the main slope movement processes providing organic debris to reservoirs.

Debris Slides

Debris slides are restricted to slopes in excess of 20° and their frequency of occurrence increases with slope up to 45°, or so, above which the slopes are often devoid of soils and slides fail to develop. Debris slides move from a few to ten thousand m^3 of soil and debris which is often deposited at changes of slope, on benches or on the valley floor. The larger slides have a much greater probability of reaching a reservoir (Rood 1984). Natural rates of failure vary from region to region in British Columbia (Table 1).

Logging of steep slopes affects debris sliding. Within the clearcut deterioration of the root mat and yarding disturbance of soils increase the frequency of sliding by factors ranging from 2 to 40 times (Table 1). For slides originating within logged blocks the amount of organic debris entrained may be small, consisting of slash, stumps and broken trunks. However, those failures that enter the surrounding forest will entrain large volumes of organic debris.

Both clearcutting and road construction accelerate the volume of sliding relative to that observed in forested terrain. The accelerations are variable, but a modest acceleration is often observed in the clearcut areas as a result of root decomposition, leading to reduced slope stability, and a greater acceleration is observed from the road prism as a result of fill slope failure, culvert blockages and drainage diversion.

Table 1: Regional rates of debris sliding and torrenting for steeplands[1]

Region	Annual Forested Rate[2]		Acceleration by Logging[3]	
	Debris Slides (ha/km^2/yr)	Debris Torrents (m^3/km^2/yr)	Clearcutting	Road
Coast Mtns - Queen Charlottes - Western Van. Island	0.004 - 0.012 0.02 - 0.10 0.01 - 0.05	5 - 50 10 - 100 5 - 50	2 20 - 40 10 - 20	25 - 125 100 - 300 100 - 300
Interior	<0.001	<1	2	50 - 100
Northern Mtns	0.002 - 0.005	5 - 25	5	75 - 150
Rocky Mtns	.002 - 0.005	5 - 25	1 - 3	75 - 150
Columbia Mtns	0.001 - 0.004	1 - 10	1 - 3	75 - 150
Alberta Plateau	<0.001	<1	1 - 3	50 - 100

1. Steeplands include all forested slopes greater than 20°.
2. Long-term mean annual rates. Quoted rates are derived from the summary in Sidle et al (1985), Rood 1984; or Rollerson 1984.
3. Accelerations are the increase in the forested rates that apply to the area that is clearcut or the area covered by roads within the harvested blocks. Quoted accelerations are derived from the references in the above footnote or estimated.

Debris Torrents

Debris torrents typically originate in small, steep creeks or gullies, with abundant debris, and are triggered by small debris slides, bank failures, rock falls or liquefaction of bed material. They may travel up to several km, scouring the channel as they proceed, and transport up to 100,000 m^3 of soil and organic debris though most involve less than 10,000 m^3.

Coarse debris torrents generally deposit on slopes of 14° or less, or where the flow becomes unconfined (Hungr et al 1984), or they often jam in less steep stream reaches, or deposit on steep fans or cones. Some of the torrents reach major stream valleys. Flooding of reservoirs may drown footslopes and fans and increase the probability of debris torrents directly entering the reservoir.

The proportion of organic debris in the deposit is seldom measured but may be large. Along the maritime coast of British Columbia the majority of the torrent volume may consist of organic debris. Much lower organic volumes are found in natural debris torrents in the Coast Mountains of B.C. and those that re-occur in structurally-controlled sites in the Coast Mountains have negligible organic content.

Logging of steep slopes also increases the frequency of debris torrenting, though there are few studies. Frequencies of torrenting increases from 5 to 20 times, typically because the frequency of triggering landslides increases, and the torrents increase in size and become more mobile. In the

Queen Charlotte Islands, flows originating in logged blocks may have more smaller debris -- logging slash, etc -- and less overall organic debris particularly if the torrent travels mostly through clearcut areas where gullies have been carefully treated during logging.

Estimation of Supply to Reservoirs

Table 1 summarizes annual rates of soil disturbance by debris slides for steeplands in the various regions of British Columbia. Average annual contributions of organic debris from debris slides are calculated from the area eroded by debris sliding (Table 1 or interpreted from aerial photographs), the area of steepland on the reservoir slopes, the volume of forest cover, and a percentage to represent the portion of the debris entering the reservoir. This factor will be less than 100% under all circumstances and will approach zero where the steeplands do not extend downslope to typical reservoir levels. Rood (1984) found that roughly 40% of the mobilized slide volume reached valley bottoms and entered streams in the Queen Charlotte Islands.

Contributions from clearcut areas and roads are calculated by the same procedure but use an accelerated rate of soil erosion (Table 1 or interpreted from aerial photographs). Three factors affect the calculation of debris supply from harvested terrain. Accelerated rates only apply to clearcuts for five to fifteen years after logging and there is tremendous variability from clearcut to clearcut. Second, the slides in clearcuts are smaller than those in forested terrain, less likely to reach the valley bottom and contribute, proportionately, less of the total slide volume to the reservoir. Third, in some instances, very little large organic debris is contributed to the reservoir particularly if a thorough clearing of logging slash from gullies and slopes is completed at the end of harvesting.

Landslide inventories are occasionally collected following major storms. These storm inventories indicate that much of the volume of sliding and torrenting observed in long-term studies results from a few, large storms, and it may be reasonable to assume that medium to large storms supply several tens of years worth of the average annual yield.

The annual volume transport rate for debris torrent is selected from Table 1 or estimated from measurements on aerial photographs and the contribution to a reservoir is calculated from the volume transport rate, the forested steepland area along the reservoir margin and the proportion of organic material in the debris torrents. The proportion of organic debris is best estimated from aerial photographs or field inspection though 25% may be assumed for initial calculations. A similar procedure is used for calculating contributions from clearcut and roaded terrain.

In steeplands, both clearcutting and road building increase the frequency of debris torrents. However, few studies have examined the effect of harvesting on acceleration of debris torrenting and the accelerations appropriate for debris sliding can be applied as an estimate of the increase in debris flows. This is reasonable on the maritime coast of British Columbia, where the increase in debris torrenting is directly related to the increase in the number of triggering events. However, in the interior of British Columbia, drainage diversion along roads is the main source of debris torrents and accelerations may depend mostly on road-building practices.

Summary and Conclusions

Our approach to estimating organic debris inflow to B.C. Hydro reservoirs was based on an organic debris budget, which defined the additions to the volume stored in the reservoir from shoreline erosion and blowdown, landsliding along the reservoir slopes and transport by tributary streams. The supply of large organic debris (which includes all stems and branches over 10 cm in diameter) was evaluated because it is this debris that has the potential to block or damage discharge structures. Inflow quantities during large storms were the main focus as blockage or damage during these storms has the greatest potential to affect dam safety.

The approach described is flexible and may be used in the office, with maps and reports, to provide rapid estimates for reservoirs where exact debris inflow quantities are not of great importance. These estimates are suitable for assessing the relative importance of various sources of debris and for ranking or priorizing debris contributions to reservoirs in the B.C. Hydro system. The approach may also be used with field observations and measurements for a specific reservoir to prepare improved estimates of the quantity of inflow debris. Ultimately, accurate inflow estimates for reservoirs, will require capture and estimation of quantities of debris entering the reservoir from tributaries in order to calibrate the approach.

References

Beaudry, P.G., D.L. Hogan and J.W. Schwab. 1990. Hydrologic and geomorphic considerations for silvicultural investment on the Lower Skeena River floodplain. Canada-British Columbia Forest Resource Development Agreement Report 122. 41 pp.

Fiksdahl, A.J. 1974. A landslide survey of Stequaleko Creek watershed. Supplement to Final Report FRI-UW-7404. Fisheries Research Institute, University of Washington. Seattle, Washington. 8 pp.

Harmon, M.E., J.F. Franklin and F.J. Swanson. 1986. Ecology of coarse

woody debris in temperate ecosystems. **in** Macfadyen, A and E.D. Ford (eds). Advances in ecological research. Academic Press, New York. Volume 15: 133-302.

Hogan, D.L. 1986. Channel morphology of unlogged, logged and debris torrented streams in the Queen Charlotte Islands. B.C. Ministry of Forests and Lands. Land Management Report No. 49. 94 pp.

Hungr, O., G.C. Morgan and R. Kellerhals. 1984. Quantitative analysis of debris torrent hazards for design of remedial measures. Can. Geotechnical Journal 21: 806-813.

Knighton, A.D. 1987. River channel adjustment - the downstream dimension. **in** K. Richards (ed). River channels: environment and process. The Inst. of Brit. Geog. Spec. Pub. Ser. 18. pp. 95-128.

Ministry of Forests. 1992. Biogeoclimatic zones of British Columbia 1992 (Scale 1:2,000,000). B.C. Ministry of Forests. Research Branch.

Morgan, G.C. 1982. An approach to predicting and evaluating the shoreline performance of man made lakes. Proc. IV Congress of the Int. Assn. of Engineering Geology, New Delhi. VII.47-VII.55.

Neill, C.R. 1971. River bed transport related to meander migration rates. ASCE Proceedings. J of the Waterways Div. 97(WW4): 783-786.

Nielsen, N.M. 1992. Reservoir debris - safety, economic and environmental considerations. Hydropower 92. Lillehammer, Norway.

Rollerson, T.P. 1984. Terrain stability study - T.F.L. 44. MacMillan Bloedel Ltd. Woodlands Services Division, Nanaimo, B.C. 63 pp.

Rood, K.M. 1984. An aerial photograph inventory of the frequency and yield of mass wasting on the Queen Charlotte Islands. B.C. Ministry of Forests Land Management Report No. 34. 55 pp.

Sidle, R.C., A.J. Pearce, C.L. O'Loughlin. 1985. Hillslope stability and land use. Water Resource Monograph 11. American Geophysical Union. Washington, D.C. 140 pp.

Varnes, D.J. 1978. Slope movement types and processes. **in** Landslides: analysis and control. Special Report 176, Transport Research Board, National Academy of Science, Washington, D.C. pp. 11-33.

Sediment Transport Assessment in the Old River Control Area of the Lower Mississippi River (USA)

Sultan Alam[1], Cecil W. Soileau[2] and Ralph L. Laukhuff[3]

Abstract

Water and sediment diversion from the Mississippi River at the Old River Control (ORC) has been an important aspect of the Lower Mississippi flood control system for the U.S. Army Corps of Engineers (ACOE) since the early sixties.

The construction of the Sidney A. Murray Jr. Hydroelectric station has significantly modified the operation of the flow diversion structures at ORC and consequently the sediment transport process. To assess the actual sediment transport conditions the Owner of the Hydroelectric Station in agreement with the ACOE New Orleans District has undertaken a very ambitious sediment monitoring program since February, 1991.

In the field of sediment transport survey, such detailed data collection and quantitative sediment transport assessment has never been done before. Enough data has now been collected, analysis of which enables us to draw some interesting conclusions.

ACOE New Orleans District Sediment Program

The Mississippi River drains 41 percent of the contiguous 48 states of the United States and a small portion of Canada, an area of 1¼ million square miles (Figure 1).

The Mississippi River enters the New Orleans District just above Old River at Black Hawk, La. This is at River Mile 324 above Head of Passes of the Mississippi River Delta (Figure 2). The Sydney A. Murray, Jr. Hydroelectric Station is located at River Mile 316.

Flow is diverted through the Old River Control Complex, consisting of the hydropower plant and two ACOE structures, such that an amount equal to 70 percent of the combined flow of the Mississippi and Red Rivers (Total Latitude Flow: TLF) continues down the lower Mississippi and an amount equal to 30 percent passes down the Atchafalaya River (Figure 3). The ACOE New Orleans District currently collects suspended sediment data on the Mississippi, Atchafalaya, Old River Outflow Channel and Red Rivers.

(1) Chief Engineer, Hydraulics, SOGREAH
(2) Chief Hydraulics and Hydrology, US ACOE New Orleans District
(3) Station Manager, Sidney A. Murray Jr Hydroelectric Station, Louisiana Hydroelectric

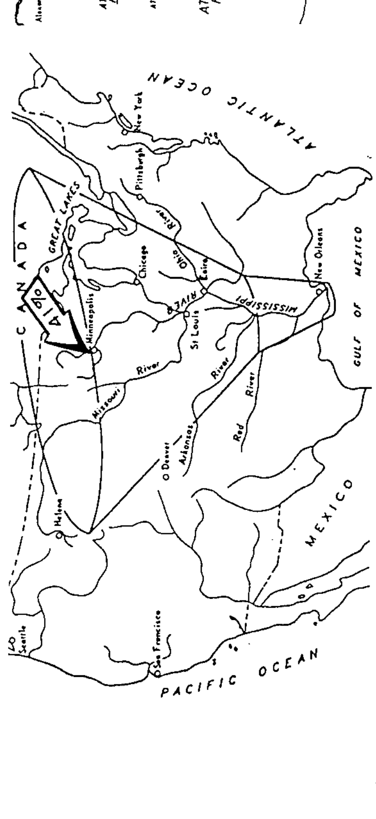


Fig. 1 THE MISSISSIPPI RIVER CARRIES WATER FROM 41 PERCENT OF THE 48 CONTIGUOUS STATES OF THE UNITED STATES AN AREA OF 1.25 MILLION SQUARE MILES

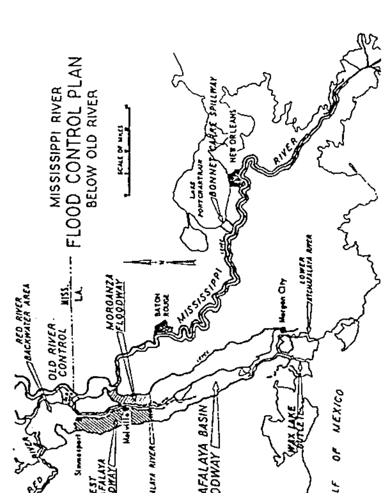


Fig. 2 MISSISSIPPI RIVER FLOOD CONTROL PLAN

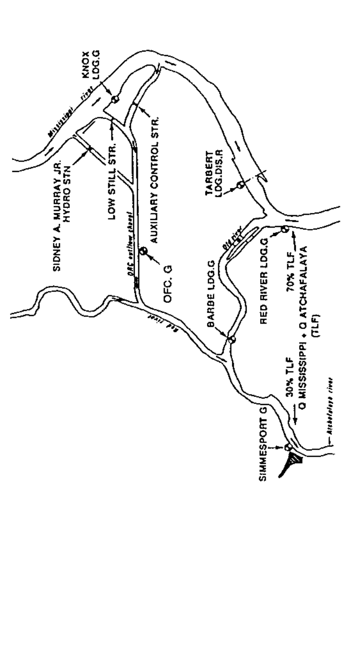


Fig. 3 FLOW DIVERSION ARRANGEMENT AT THE OLD RIVER CONTROL

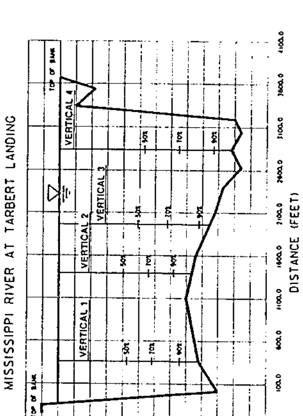


Fig. 4 TYPICAL SEDIMENT SAMPLING POINTS

The two major stations (and stations with the longest period of data collection) are located on the Mississippi River at Tarbert Landing, which is just above Old River at River Mile 306, and on the Atchafalaya River at Simmesport at River Mile 4 downstream of Old River. Another important station is located on the Old River Outflow Channel downstream from all of the diversion structures. The purpose of the sampling program is to determine the amount and characteristics of sediment being transported by the major rivers in the district. It is also needed to insure that the proper distribution of sediment is achieved at Old River. The Mississippi River is attempting to change courses at Old River and take the shorter route of the Atchafalaya to the Gulf of Mexico, and one of the major goals of the ACOE is to keep it on its present course. If too little sediment is diverted through the Old River Complex, the Mississippi River will potentially lose channel capacity in the reach below; and, this will aid the river in its effort to change its course.

Sediment Sampling in the Field

On the Mississippi River at Tarbert Landing, point sediment samples are taken twice a month, using a P-63 sampler on four verticals at three points on each vertical. The points are at 50 percent, 70 percent, and 90 percent of the depth (Figure 4). Similar sampling is also done in the Old River Outflow Channel and at Simmesport on the Atchafalaya River. The samples are subjected to standard lab analysis.

The concentration at the points are discharge-weighted to obtain a composite concentration for that observation. At the end of the year, a sediment discharge hydrograph is created by computer using the year's flow hydrograph in conjunction with the sediment observations for the year. The computer program uses the shape of the water discharge hydrograph to create the sediment discharge hydrograph, but forces the sediment discharge hydrograph to intersect all sediment observation data points. The daily values composing the sediment hydrograph are summed to produce the annual sediment load.

A bed sample is also taken at each vertical, using a BM-54 sampler. A composite of the bed samples for the four verticals is sent to the lab, where a seive analysis is performed and a bed material gradation curve produced. Some characteristics of the sediment transport at Tarbert Landing and ORC Outflow Channel for water years 1987-1991 are shown on tables 1 & 2. A great amount of effort has been expended at Old River to try to keep the amount of sediment diverted as large as possible. Prior to construction of the Diversion Structures, extensive model tests were undertaken to determine the configuration of the structures and inflow channels which would be conducive to maximum sediment diversion. Model tests by the ACOE indicated that as much as 65 percent of the bedload of the Mississippi River could be diverted. These predictions were borne out. Soon after the operation plan was put in place in 1986, the conveyance of the Mississippi River channel in the reach below Old River began to improve.

However, since May 1990 Sidney A. Murray Jr. Hydroelectric Station has been in operation and significantly changed the flow and sediment distribution at the Old River Complex. Before the inception of the hydro station, the quantity of water diverted from the Mississippi River had the following distribution:

For Knox Landing stages between 70 ft and 30 ft, 37 to 40% of the flow passed through the Auxiliary Structure and the remaining 63 to 60% passed through the Low Sill Structure.

For Knox Landing stages between 30 ft and 10 ft, 40 to 65% of the flow passed through the Auxiliary Structure and the remainng 60 to 35% passed through the Low Sill Structure.

Since the completion of the hydro station, the ACOE has agreed to allow the hydro station (when possible) to take all the available discharge (to the extent of plant saturation) for power generation, and the remaining excess discharge to be diverted through the Auxiliary Structure or the Low Sill Structure which is used only on rare occasions and for very short periods. The 1991 hydrological year, excepting the month of December, was a very low flow year for the Mississippi River and a very high flow year for the Red River. Consequently the proportion of water diverted from the Mississippi River to assure the aforementioned 70-30 split was relatively small. The actual percentage of flow diverted through the various structures were the following:

Hydro Station (PPS)	59%
Auxiliary Structure (AUX)	36%
Low Sill Structure (LSS)	5%

Daily Sediment Transport Assessment Program

Considering the paramount importance of sediment diversion at the ORC it was essential to assess as precisely as possible the impact of this major change due to the flow diversion through the hydro station on the pre-hydro sediment transfer balance. For this reason the Owner of the hydro station, Louisiana Hydroelectric, has undertaken two separate study programs:

- Studies on the existing movable bed physical model at WES, Vicksburg, of the possible structural arrangements and operating procedures to achieve the required sediment diversion at the ORC.

- An extensive sediment monitoring program at all the ORC structures since February 1991.

The monitoring program in progress is unique in the annals of prototype sediment transport survey, because for the first time it is possible to assess with great accuracy the total daily sediment transport in the Mississippi River and at the various ORC structures (Figure 3). Three different monitoring procedures are being used:

- Daily sediment sampling at the power plant draft tube outlet and in the energy dissipation basins of the Auxiliary and Low Sill Structures.

- Continuous recording of the sediment concentration at all the above-mentioned structures by using optical scanners.

- Regular hydrographic survey of the Mississippi River and the inflow and outflow channels to and from the various structures involved in the sediment diversion process.

Figure 5 shows schematically the sediment sampling and continuous recording arrangements. Water is pumped continuously from areas where very high turbulence maintains the total sediment load (including coarse sand) in suspension (Plate 1):

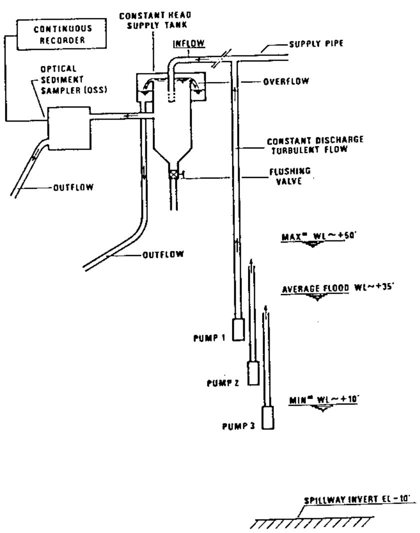


Fig. 5 SCHEMATIC ARRANGEMENT OF THE SEDIMENT SAMPLER INSTALLATION

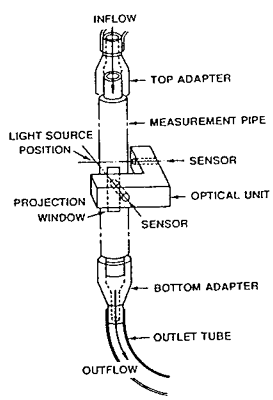


Fig. 6 OPTICAL SEDIMENT SAMPLER

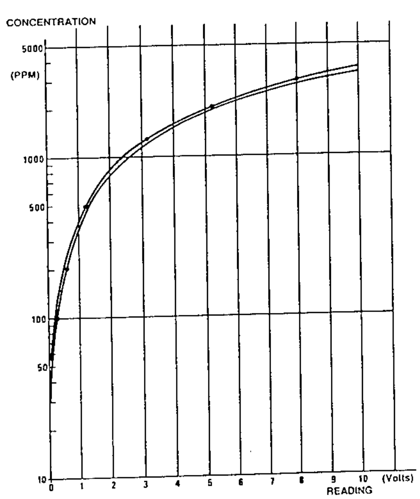


Fig. 7 TYPICAL CALIBRATION CURVE OF THE OPTICAL SEDIMENT SAMPLER

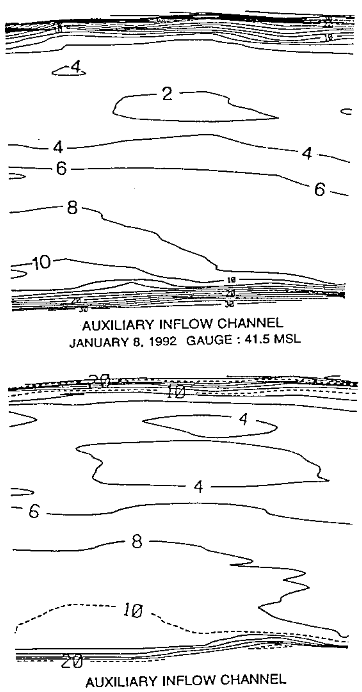


Fig. 8 HYDROGRAPHIC SURVEY OF SEDIMENT ACCUMULATION IN THE AUXILIARY INFLOW CHANNEL BETWEEN JAN AND APR. 1992

- For the power plant, it is at the draft tube summit 40 ft downstream of the turbine runner chamber. Three out of eight units are equipped with such pumps.

- For the Auxiliary and Low Sill Structures three well distributed verticals across the structures, each equipped with three pumps at three different levels (9 pumps on each structure - Plate 1).

A commutator arrangement ensures 20 minute constant discharge from each pump, which is then conveyed through a delivery pipe to a constant-head tower (guaranteeing required turbulence to maintain the total sediment load always in suspension) assuring constant velocity flow through the optical scanner.

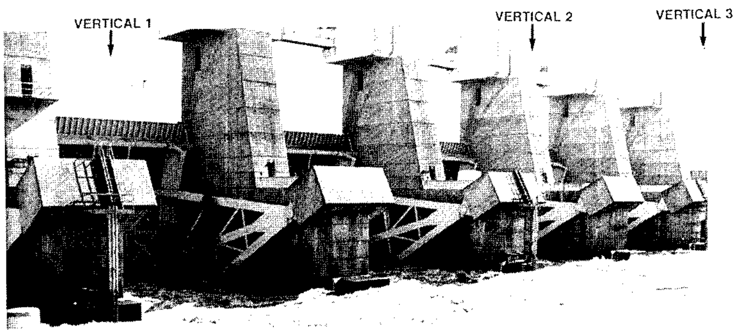
Daily sediment samples are collected at the exit of the optical scanner outflow, and are then analysed by using the standard ACOE laboratory practice. The window of the optical scanner (Figure 6) is cleaned every day so that its reference transparency remains constant. Figure 7 shows a typical calibration curve. The correlation between the actual measured total sediment concentration to that obtained from the reading of the optical scanner is good. However, with respect to the sand content, a major difference between normal turbulent river or channel flow and flushing operation is observed; with experience it should be possible to estimate the total sand content fairly accurately by knowing the operating conditions and time of the year.

Based on the continuous recording, it is possible to observe major variations in total sediment concentrations (compared to that of the daily samples) during a reference 24 hour period and if required make necessary adjustments. However, the most interesting aspect of the installation at the ORC is the possibility of taking samples containing the total sediment load every day, any time of the year, without much difficulty. This fact alone improves the precision and reliability of the sediment assessment process.

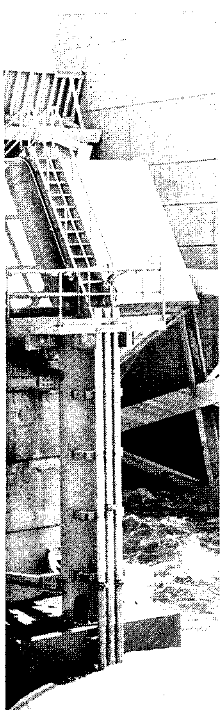
Figures 9 and 10 show the daily variations in the total sediment concentration and the corresponding total sand content as a function of the Mississippi River discharge and the flow diverted through the PPS and the AUX.

It can be seen that during the first half of the rising flood hydrograph of the Mississippi River the sediment concentration at the PPS increases sharply and attains a maximum of 1380 ppm on December 8, 1991 (Figure 9), and then even though the Mississippi River discharge continues to increase, the sediment concentration goes down gradually. At the peak discharge of 1 000 000 cfs the concentration is only 260 ppm. We feel that in this case the total sediment concentration measured at the PPS also reflects the sediment concentration of the Mississippi River, because the PPS intake channel is not prone to any erosion or deposition. This figure also shows that periodic point sampling may not always give the most representative sediment concentration values. Analysis of cumulative sediment data (February to October 1991) has also revealed that the total sand content is about 15% of the total sediment load. During the same period the total sediment concentration passing through AUX is much less and its sand content almost negligible. The AUX intake channel at this stage serves as a sediment storage area and captures all particle sizes except the very fine ones. This has been verified by periodic hydrographic surveys (Figure 8).

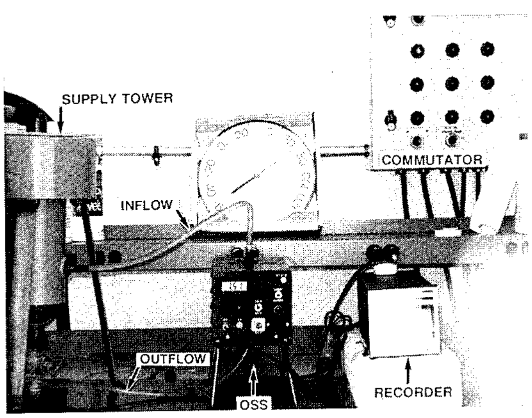
PLATE 1



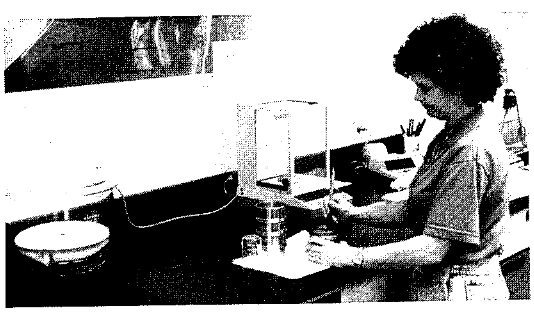
GENERAL ARRANGEMENT OF THE SAMPLING VERTICALS AT THE AUXILIARY STRUCTURE



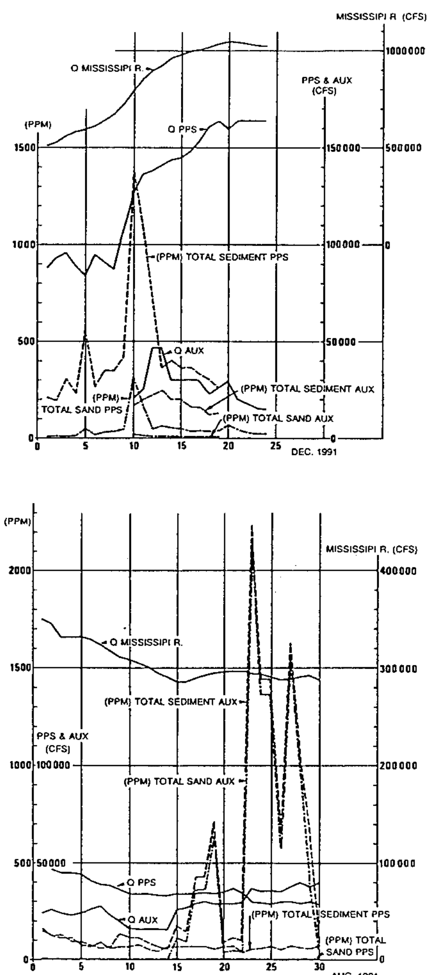
DETAILS OF THE PUMPING INSTALATION



CONTINUOUS SEDIMENT SAMPLING INSTALATION



SIEVE ANALYSIS



VARIATION OF SEDIMENT CONCENTRATION AS A FUNCTION OF MISSISSIPPI, POWER PLANT (PPS) AND AUXILIARY STUCTURE (AUX) FLOWS

Fig. 9 and 10 DAILY SEDIMENT CONCENTRATION

The whole process is reversed during the low flow period when the PPS as well as the Mississippi River sediment load is very small, but due to the low river stage even for AUX discharges between 30 000 and 40 000 cfs its inflow channel starts flushing out the sediments accumulated during the high water stages. A combination of falling Mississippi River stage and rising AUX flow produces an extremely sharp increase in the sediment concentration, and on August 23, 1991 (Figure 10) it is 2 230 ppm, which is probably the highest ever measured in the ORC area. It is also interesting to note that this sediment load is entirely composed of sand. The sediment flushing process tapers off after about 15 days and the total concentration drops down to only 120 ppm on August 30, 1991, although the flow is maintained or even slightly increased. This also confirms that the daily sediment transport measurement can optimize both the sediment flushing operation at AUX and power generation at PPS, during low flow periods of the Mississippi River.

Total weights of different categories of sediment diverted at ORC during the 9 month period (February-October 1991) are as follows:

Total sediment .... 15 528 756 tons
Total sand (> 0.06 mm) .... 2 306 603 tons
Coarse sand (> 0.125 mm) .... 1 779 772 tons

It can be seen that the total sand is 15% of the total sediment load and the corresponding proportion of coarse sand is 11.5%. The corresponding suspended sediment distribution at the various ORC structures are:

PPS 62%
AUX 32%
LSS 6%

This is almost identical to the % of flow diversion (mentioned earlier). However, the % of the total sand and coarse sand diversion at the various structures are very different, as can be seen in the table below:

Structure	% Total Sand	% Coarse Sand
PPS	22.0	13.9
AUX	72.9	82.1
LSS	4.9	4.0

Comparison of Sediment Load

Comparison of a few sediment loads measured using conventional point sampling methods and the daily sampling method as practiced at the ORC structures shows significant differences:

Table 1

MISSISSIPPI RIVER SEDIMENT LOAD						
Item (water years 1987-1991)	Point sampling			Daily sampling		
	Q_{Missi} at Tarbert LDG. (cfs)	Suspended sediment concentration (PPM)	Tons/day (10^6)	Q_{Miss} at Tarbert LDG. (cfs)	Total sediment concentration (PPM)	Tons/day (10^6)
Observed maximum (10-16-86)	737 000	559[(1)]	1.111	646 500*	1 380	2.405
Observed minimum (08-15-88)	120 000	57	0.018	208 000**	43	0.024
Average annual mean	535 800	228	0.329			

* 12-10-91.
** 09-18-91.
(1) d_{50} = 0.34 mm (sand)

Table 2

ORC OUTFLOW CHANNEL SEDIMENT LOAD								
Item (water years 1987-1991)	Point sampling			Daily sampling				
	Discharge outflow channel (cfs)	Suspended sediment concentration (PPM)	Tons/day (10^6)	Discharge (cfs)		Total sediment concentration (PPM)		Tons/day (10^6) Total
				PPS	AUX	PPS	AUX	
Observed maximum (04-06-88)	99 200	432	0.115	37 000*	30 000*	60	2 230	0.087
Observed maximum (01-24-89)	250 000	396	0.267	138 000**	20 000**	1 380	170	0.522
Observed minimum (10-11-91)	42 600	40	0.005	33 000***	16 000***	60	60	0.008
Average annual mean	138 860	213	0.080					

* (08-23-91).
** (12-10-91).
*** (08-14-91).

The table below shows the comparison of sediment data for ORC Outflow Channel for the water year 1992:

Suspended sediment	Point sampling (ACOE)	Daily sampling (Louisiana Hydro)
Total load	19.6×10^6 tons	22.7×10^6 tons
Total sand	2.4×10^6 tons	1.7×10^6 tons

Conclusion

The daily sediment sampling program initiated at the ORC has revealed many interesting facets of the sediment transport process and analysis of data collected since February, 1991 enables us to draw the following conclusions:

- Total suspended sediment transferred through the various structures is proportional to their water discharge.

- The maximum rise in sediment concentration occurs during the early stage of the rising flood.

- Both total suspended load and total sand diverted at ORC may vary widely from year to year.

- For a given flow distribution the Auxiliary Structure diverts 5 times more sand and 10 times more coarse sand than the power plant. However a permanent increase in the Auxiliary Structure discharge will decrease its efficiency and the overall increase in sand transport will not be proportional.

- The daily sampling procedure helps to enhance and optimize both sediment transfer and power generation.

- The total suspended sediment load assessed by point sampling method does not include the corresponding bed load, comparatively daily sampling from areas with extremely high turbulence include the local instantaneous bed load.

- The daily sampling procedure may be of great interest for projects where precise quantitative total sediment load assessment is required.

Hydro-electric Generator Ozone Monitoring

D.E. Franklin[1], B.C. Pollock[1], J. Laakso[2]

Abstract

The presence of ozone in relatively high concentrations provides an indication of faults within generator winding, insulation and brush gear systems. In addition to being a health and safety concern, if the concentration of ozone is monitored on-line over time, it could serve as a good indicator of machine condition, and identify the rate of developing fault problems. Damage of other plant equipment (i.e. fire hoses, connectors, rubbers, etc.) from ozone attack is also a growing concern. Ozone Monitoring Systems could be very useful both as a predictive maintenance tool, and a safety measure for plant personnel. This paper describes an on-line Ozone Monitoring System which was developed by Powertech Labs through the support of funding provided by B.C. Hydro and the Canadian Electrical Association (CEA). The developed system is scheduled for installation on two 175 MW generating units at B.C. Hydro's Peace Canyon Generating Plant in February 1993.

Introduction

Powertech Labs Inc. is currently involved in the development of a comprehensive Hydro-electric Machine Condition Monitoring initiative. An ozone measurement system is being developed with an approach towards incorporating flexibility, so that the device will be fully utilized within a future more encompassing machine or plant monitoring system. As well, the ozone monitoring system is being developed for stand-alone operation. This enables the system to be set-up for any plant, ensuring technology transfer to hydro-electric generation areas.

---

[1]Powertech Labs Inc., 12388 88th Avenue, Surrey, B.C., Canada, V3W 7R7
[2]B.C. Hydro, E07, 6911 Southpoint Drive, Burnaby, B.C., Canada, V3N 4X8

Record

Powertech Laboratory personnel have been involved with the periodic monitoring of ozone levels in B.C. Hydro generating stations for the past 24 years. The measurements performed, primarily for industrial hygiene purposes, are measured using a portable ozone analyzer equipped with a chemiluminescent detector. The magnitude of field ozone measurements, particularly the low levels, have also been checked occasionally using a Draeger tube. In recent years, the industrial trend for ozone monitoring has been more toward an active measurement technique where air samples are drawn on a continuous basis. This approach better profiles ozone generation as a function of time, thus eliminating the errors possible with snap-shot monitoring.

Ozone is a potent, toxic, colourless gas with a distinctive, pungent odour. The respiratory system is the primary site of attack; however, ozone may also affect the blood, central nervous, endocrine, and reproductive systems. It can cause damage very quickly, even is very small doses. The odour is barely detectable at 0.015 parts per million (ppm). At 1.0 ppm the odour is strong, sulphur-like, and disagreeable. A limit of 0.10 ppm has been established by British Columbia's Workers Compensation Board for allowable exposure concentrations over an 8 hour period.

Ozone (O_3), also known as triatomic oxygen, is produced by the decomposition of molecular oxygen (O_2) in the atmosphere.

$$O + O_2 \longrightarrow O_3$$
(Free reactive oxygen atom combines with oxygen to form Ozone)

It is generated from many sources including the operation of high voltage electrical equipment. In Air-Cooled Generators, slot discharge, arcing brushes, and any other form of corona activity produces ozone.

$$O_2 + \text{high energy electron} \longrightarrow 2O + \text{low energy electron}$$
(Corona Discharge produces reactive oxygen atoms)

Passive Ozone Testing

Snap-shot ozone concentration measurements taken within B.C. Hydro generator enclosures have shown a wide variation in the generators tested to date. Within these generators, measurements have ranged from as little as 6 ppb up to 260 ppb. Measurements taken at the turbine level below the generators have shown some of the highest ozone levels (averaged at 37 ppb), whereas concentrations up to 25 ppb have been found in the cooling air to the generators. Most of the readings outside the generators have been much lower, generally less than 10 ppb.

Snap-shot ozone measurements were taken inside each generator enclosure at the Peace Canyon Generating Plant (14 February 1992). The following levels were found:

G1	234 ppb
G2	244 ppb
G3	168 ppb
G4	260 ppb

Despite the low concentrations generally found outside the generators, there have been some indications that much higher ozone levels may occur. At one generator station, relatively new lineman's gloves have shown cracking typical of that produced by ozone attack after very little time in use. Failure of a fire protection hose during a routine safety test is also recorded as being caused, at least in part, by ozone attack. On another unit at a different generating plant ozone has been detected by smell, suggesting the presence of high ozone concentrations.

Development of on On-line Ozone Monitoring System

Powertech Labs is developing and will implement an ozone monitoring system which will maximize the use of available commercial technology. The system will be capable of monitoring multiple machines with the ozone level results available both on the plant floor, and remotely, for trending and evaluation purposes. Commercially available hardware and software is being utilized wherever possible.

Measurement Points

Measurement points identified from experience as the most likely ozone concentration sites are:

Inside Generator	
behind heat exchangers (ozone from stator windings)	(4 points)
near brush gear (i.e. in brush gear cubicle)	(1 point)
generator pit	(1 point)
Outside Generator	
plant floor (between generators)	(1 point)
turbine pit	(1 point)
	16 points

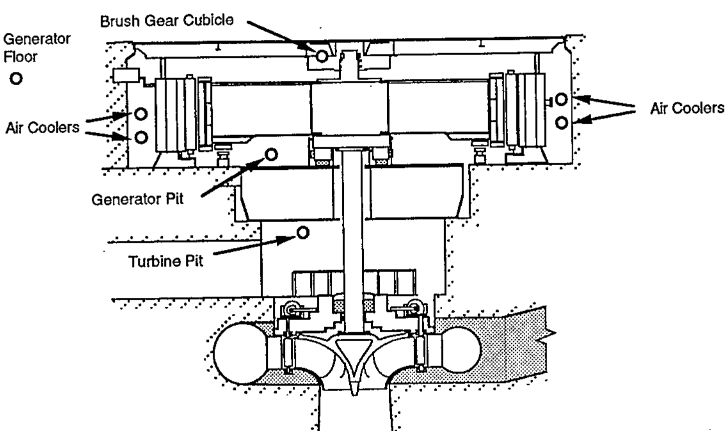


Figure 1: Ozone Monitoring Locations for Units No. 1 & 3 at BC Hydro's Peace Canyon Generating Plant

The system is initially configured with 8 measurement points per machine. This number could be increased or reduced accordingly once preliminary site measurements are confirmed (i.e. the system will not be limited in the number of measurement points possible).

Gas Sampling and Multiplexing

An in-depth evaluation of commercially available gas multiplexing systems revealed no system that would meet the full requirements of the program. Consequently, the configuration of modular, commercially available valving and electronics has been selected as the best alternative. This multiplexing unit is rack mounted with the analyzer near the generator. With this arrangement, both the ozone ppb levels and channel number will be displayed on the plant floor beside the generator. Teflon components were required for all gas handling operations as many commonly used materials will affect the measured ozone levels.

To ensure reliable ozone measurements, the air samples will be pumped continuously through teflon tubes to the ozone measurement device. The samples will also be appropriately filtered at the intake end to remove possible contaminants such as brake dust.

The continuously flowing samples are multiplexed or scanned and individually analyzed in approximately three minutes. The scan time to complete an analysis of samples taken from the 8 areas on one generator is approximately 30 minutes. It is felt that scanning different points on a

continuous basis will provide the greatest benefits while reducing the cost of implementing multiple instruments for analysis. It has been unnecessary to develop sensors since commercial products are available. Some systems offer sensors at various locations, while others have one detector that multiplex samples from a selection of channels. Powertech's approach is to use the latter sensor which greatly simplifies calibration of the unit and ensures accurate repeatable results, while reducing costs.

Correlation of ozone measurements to other indicators of machine condition such as partial discharge may or may not be appropriate. Measurements taken in the ring bus, or heat exchanger areas will not necessarily provide correlation to partial discharge occurring in the stator. Consideration will be given to accommodate relationships between ozone and other measurements to assess machine condition; however, it is expected that these relationships will have to be developed on a machine specific basis.

Summary of Commercially Available Analyzing Technology

A recent review of manufacturers of low level ozone monitoring equipment and ozone generators for calibration indicated that three different types of sensing technologies are currently employed:

1. Membrane electrolysis detectors
2. Chemiluminescence detectors
3. Ultraviolet absorbancy detectors

These sensors are all reported to be capable of detecting 1 part per billion (ppb) of ozone; however, each system has limitations.

Membrane Electrolysis Detectors

Currently, membrane electrolysis instrumentation is the only system which employs individual remote sensors, either diffusion or suction devices, which transmit readings to a monitor located up to a mile away from the sensor. Air is swept across a membrane which selectively allows the ozone to permeate onto the surface of a thin film electrode where it causes an oxidation reduction reaction and produces a current which is directly proportional to the partial pressure of the gas. The detector is selective (chlorine is the only reported interferent), however the suitability of this system is questionable since the repeatability is specified as being within ±5%. This system also has the disadvantage of requiring that each sensor be individually calibrated. In a long-term monitoring application, the logistics of calibrating a large number of individual sensors would be unacceptable.

The remaining two methods (Ultraviolet absorbancy and Chemiluminescent) employ a central analyzer and therefore require sampling lines and a pump. The gas flow in the lines could be continuous

and would be sequentially sampled as required. The main advantage is that only one analyzer would have to be calibrated which could easily be accomplished on-line by having a built-in ozone generator capable of delivering known concentrations of ozone located next to the analyzer.

Chemiluminescent Detectors

The Chemiluminescent detector sensor measures the amount of light energy between 200 and 600 nm produced by the reaction between gaseous ozone and a reagent gas such as ethylene using a photomultiplier tube which converts the light energy into current and then to a voltage reading. In the presence of excess ethylene, the intensity of the light produced is proportional to the amount of ozone present. This detector is the most selective of the 3 types of detectors described and there are few potential interferents. The sensor has a response time of 2.5 seconds and the zero and span drift are reported to be 0.5% and <1.0% respectively. Almost all of the chemiluminescent sensors require a continuous flow of ethylene (about 30 cc/minute). The long-term maintainability of supplying an ethylene source for the analyzer was undesirable.

Ultraviolet Absorbancy Detectors

The Ultraviolet Absorbancy detector is essentially a spectrophotometer which measures the amount of light absorbed at a wavelength of 254 nanometres (nm) by air passing through a gas cell. The detector has a response time of about 2 minutes and is reported to be very stable (drift ±0.5% of reading) and insensitive to flow variations, but the sensor is not completely selective. Particulates and organic materials such as oil vapours also absorb light at 254 nm and would produce artificially high readings. The manufacturers have attempted to get around this problem by passing ambient air from the sampling area through a catalyst which converts the background ozone into oxygen and then measuring the absorbancy. This value, which includes the contributions from the organics and particulates in the air, is then stored in a microcomputer and serves as a reference zero. A disadvantage of this method is that erroneous readings, both positive and negative, will be obtained if the concentration of either the organics or particulates vary substantially with time unless frequent calibration checks are made. After a careful examination of the reliability and availability of commercial products, a decision was made to use the UV type detector. Filters have been implemented on all measurement lines and a calibration check is automatically performed on a daily basis.

Utilizing a device such as this will provide accurate measurements, self calibrating capability, and ease of maintenance for a permanent installation application. Two of these ozone analyzing devices will be implemented at B.C. Hydro's Peace Canyon Plant.

Two UV ozone analyzers were selected for the program:

1. Monitor Labs Model 8810 Analyzer
2. Environics Series 300 Analyzer/Generator

Laboratory Testing Program

Evaluation of Analyzer Characteristics

The two Ozone analyzers, a Monitor Labs 8810 and an Environics Series 300, were installed in our laboratory and debugged. Testing was then carried out to evaluate the operational characteristics of the instruments. Both instruments performed well and met or exceeded the manufacturer's specifications. Some of the values for zero and span drift do not appear to meet the specifications but this is probably due to the use of different methods of calculating the values. The manufacturers did not indicate how their data was calculated but they are likely based on standard deviation whereas the Powertech data represents a worst case scenario since they are the maximum and minimum values incurred during the test periods.

The 8810 unit was calibrated by the Greater Vancouver Regional District laboratory using secondary standards which are directly traceable to primary standards at Environment Canada's Ottawa laboratory. The 8810 was then used to establish a calibration graph for the Environics instrument. Both units showed excellent linearity over the expected measurement range.

Teflon Tube Testing Program (Temperature and Length Effects)

Since the length of the sampling lines in the generators will vary and some sampling points will be at elevated temperatures, experiments were also carried out to evaluate the effect of sampling line length and sampling temperature on ozone decay. The generator module in the Environics model was used to produce desired concentrations of ozone in air. The effluent gas was passed through various lengths of PFA tubing contained within an oven and then directed into the Monitor Labs 8810 analyzer. The results showed that ozone decay was independent of concentration and that the losses were very minor (2% or less) with sampling tube lengths up to 200 feet, even at temperatures up to 20°C above ambient. Tests carried out over 12 hour periods showed that the losses were constant once equilibrium was attained after about 15 minutes exposure (i.e. after line was passivated).

Particulate Contamination Considerations

It is expected that ozone decay due to reaction with particulate accumulation (especially carbon) on the teflon filters at the end of each sampling line will be more significant than line length and temperature

effects. This could cause problems in the more dusty sampling locations such as the brushgear cubicle. This effect was briefly examined by exposing both new and contaminated teflon filters to an ozone flow and comparing recovery times, i.e. the time to passivate the filter and return to 98% of the original reading. A filter holder was connected in the sampling line between the Environics generator and the Monitor Labs Analyzer in such a way that the ozone flow could be directed through the holder and filter or could be bypassed directly to the analyzer. A particulate sample was obtained from Peace Canyon Generating Station by pulling air at a rate of 2.5 litres/minute through a teflon filter for 25 Hours within the G1 generator enclosure. A second contaminated filter was prepared by adding 7 milligrams of activated carbon dust to the filter surface. The results (Table 1) indicate that problems may be encountered in sampling areas such as the brush gear cubicle depending on the amount of particulate carbon present.

Table 1

EFFECT OF CONTAMINANTS ON OZONE DEPLETION

Sample	Time (in min) to 98% recovery
New	19
G1 Filter Sample	30
Charcoal	*

* recovered to 66% after 180 minutes. Recovery to 98% unlikely as curve has flattened out.

Measures have been implemented to regularly monitor the flow characteristics through each sample line and filter to track the flow performance. Maintenance intervals for servicing line filters will also be established as part of the test program.

Monitoring System Configured for Plant Application

Computer Instrumentation for Data Acquisition and Control

A computer is used to synchronize and coordinate the gas sample multiplexers, and the ozone analyzers. An object-oriented graphical programming system (LabView) has been utilized for building all software modules. Graphical block diagrams of software are formulated using commercially available software, therefore it has not been necessary to write custom, time consuming programming code. An allotment for incorporating additional data channels for measurements other than ozone is also included. This may be very useful in the future for providing correlation of ozone data with other measured parameters, for example Complex Load (MVars) or Stator Voltage (V) may be related to ozone production and would therefore be important to correlate. The graphical

display system will also provide compatible output capabilities for integration with SCADA systems and future MCM/Plant monitoring initiatives, thus providing the potential for system wide on-line data sharing capabilities. Even with this standard integration capability, the system will remain fully functional as a complete stand-alone monitoring system with a built-in flexibility for ease of portability/customization for different plants, generators, number of channels, and monitoring objectives.

Plant Installation at BC Hydro's Peace Canyon Generating Plant

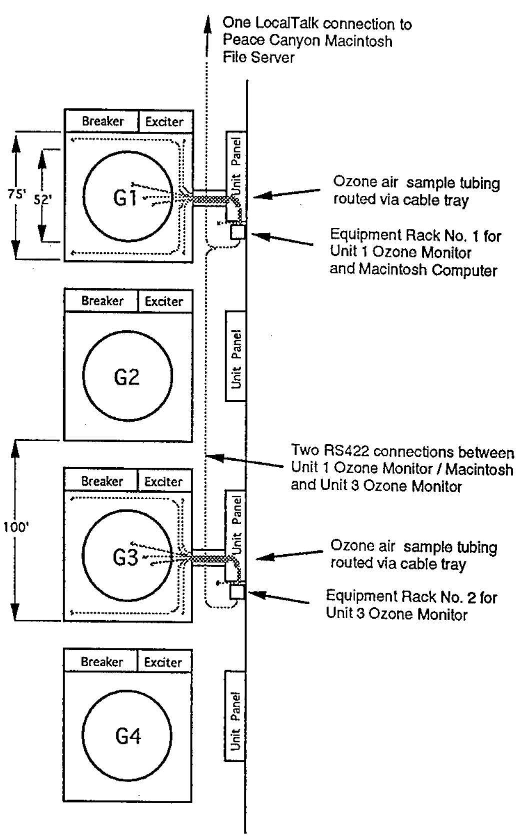


Figure 2: Plan view of Peace Canyon Plant Layout for installation of Tubing and Instrument Racks

The ozone instrument will be located in an instrumentation rack adjacent to the unit panel in the vicinity of the generator. The parts per billion (ppb) level of ozone is displayed on the instrument, and the level is sent over a communication link to a dedicated Macintosh Computer/display system (if desirable an IBM PC or other system could be utilized) also located on the plant floor in the instrument rack. The Macintosh computer is connected to the plant Local Area Network (LAN) thereby making the ozone data available to plant personnel (see Figure 2). All data measurements will be stored and will be available for trending purposes on the PC. In addition to the Ozone measurement data, diagnostic information on the performance of the Ozone analyzers, valving, and air flow system will be logged at regular intervals. On-line data will also be available for future integration with a Supervisory Control and Data Acquisition (SCADA) or Machine Condition Monitoring (MCM) system. With this arrangement the system will be capable of stand-alone operation or integration within a larger machine or plant monitoring system.

Summary

The emphasis of the approach taken is on the maximized use of commercially available technology for all aspects of the system. This includes the sample lines, sample sequencer, ozone analyzer, computer interfaces, data acquisition and display software and all industrial packaging that comes with the program. The benefit of this approach is that the finished system will have a "professional" interface and an ozone analyzer that will make use of the technology already developed and tested by others. The goal was to provide an easy to use, powerful, on-line predictive maintenance tool for plant operators.

The primary objective of the developed system is to monitor machine condition in the sense of corona discharge. Other plant equipment ozone damage may be mitigated by improved operation/maintenance of the machine by locating and repairing ozone generating areas.

A secondary benefit of the system is that Plant personnel will be protected from exposure to hazardous ozone levels. With the developed monitoring system, workers will be assured of safety before performing any work around the generating equipment or in the plant. This requirement is becoming an increasing area of concern, both from the point-of-view of a responsible employer health hygiene issue, and for compliance with Health and Safety regulations.

SUBMERGED TRASH SURVEY AT TVA DAMS

F. W. Edwards[1], D. G. Hegseth[2], and
F. R. Swearingen[3]

Abstract

The Tennessee Valley Authority is using modern electronic hydrographic survey equipment to map and study submerged, accumulated trash piles at the turbine intakes.

Text

The floating trash on Tennessee Valley Authority (TVA) lakes is continuously pulled by the water current to the generating unit intakes at TVA dams. Much of the floating trash, which consists of tree parts, wood debris, leaves, and other floating items washed from the lake shores either by rainfall, lake fluctuation, or wave action, is passed downstream through the trashways or spillways.

However, some of the floating trash waterlogs and sinks to the bottom despite the periodic maintenance described, and this submerged trash (Figure 1) accumulates near to and against the trashracks at the generating units. After several years the submerged trash can create a large pile against the trashracks and reduce the flow

---

1 - Civil Engineering Associate, Hydrographic Data Unit, Field Engineering Section, Tennessee Valley Authority
2 - Civil Engineer, Hydro Engineering Services, Tennessee Valley Authority
3 - Civil Engineer, M.ASCE, P.E., Employed by American Technical Associates, Inc., Knoxville, Tennessee

area to the turbine, increasing head loss across the trashrack. This reduces efficiency as well as increasing the structural loading on the rack. Therefore, it is important for two reasons (economics: turbine efficiency; and safety: excessive loads on the trashracks) to know the exact extent of the submerged trash at the unit intake trashracks.

Previously, TVA could only estimate the submerged trash problem at each dam by monitoring the head loss that was immediately downstream of the racks at the dams where possible, or divers could inspect the rack areas. Divers could document the elevation of the trash pile and describe the type of trash, but the size and shape of the trash pile could only be generally described by divers. When head loss is monitored and excessive loss is detected, many hours of generation may have already been run with some inefficiency. TVA sought a method of early detection of any problem caused by submerged trash. It was also desirable to have a method to estimate trash accumulation to plan and budget for trash removal prior to removal of the trashracks for painting and repair.

TVA's Mapping Services Department and the Hydrographic Data Unit routinely maps coal piles and ash ponds in the management of these operations at TVA steam plants. After meeting with representatives of the Hydrographic Unit, it was decided to conduct a test survey at TVA's Fort Loudoun Dam using their hydrographic positioning

Figure 1: Load of Trash Removed From Bottom at Trashrack

system, Hydro I[1]. The test was very successful. The topographic and isometric maps created (similar to Figures 3 and 4) clearly indicated the submerged trash piles. Cross sections, profiles, and volume tabulations were generated from the same data base (similar to Figures 5 and 6). Subsequently, turbine intakes at twenty-six TVA dams were surveyed using the hydrographic positioning system.

Hydro I is a computerized survey positioning system coupled to a digital sonar[2] unit via two dataloggers. The system works by systematically navigating a small boat (ship station-Figure 2) over the area to be surveyed while constantly tracking a target prism, mounted on the boat, with a theodolite mounted laser (shore station-Figure 2). The main datalogger is programed so that data points are collected automatically on a set interval or manually at any time the operator chooses to take an extra data point. When a data point is taken, X and Y coordinates are instantly computed by the shore station, transmitted to the ship datalogger, and stored along with the sounding (Z coordinate) from the sonar. Simultane--ously, a vertical fix mark is made on the sonar chart, at the same instant, marking the exact point the sounding is taken. Fix marks on the sonar chart allow for easy checking and editing later in the process. Hydro I also includes a navigation system to allow the boat operator

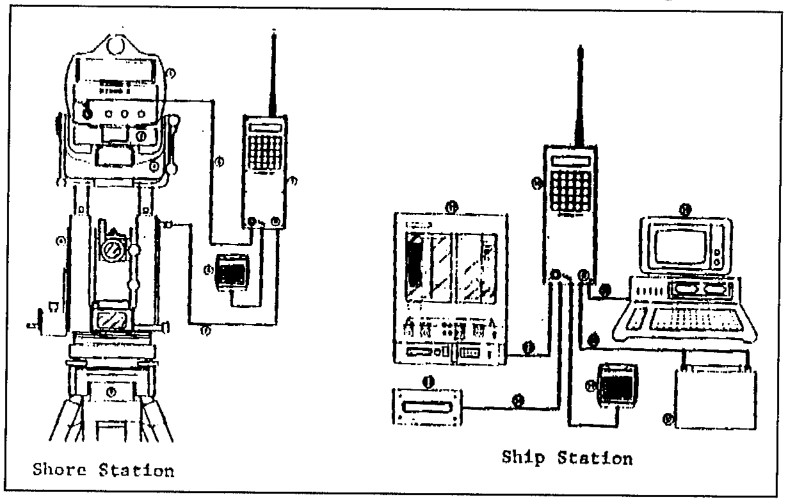


Figure 2: Survey Equipment

---

1 - Hydro I; Total Laser Survey System (TLSS); Laser Technology Inc. Englewood, Colorado.
2 - Innerspace 448, Innerspace Technology Inc., Waldwick, New Jersey.

to steer the boat over the prescribed and referenced course for the survey. The navigation system computes the prescribed boat course using coordinates transmitted from the shore station and displays course corrections with appropriate arrows on the navigator display unit.

To begin the survey, a grid system is established and referenced to the dam base line. The X axis is aligned with, or parallel to, the base line on a ninety degree azimuth clockwise from downstream in order to coordinate the survey with the map orientation. The theodolite (shore station) may be set up in any convenient known location viewing the survey area, and this can easily be done with the theodolite and laser. The theodolite must be coordinated and referenced to the 0, 0 point on the X axis and back sighted parallel to the X axis (90^0 or 270^0 must be set in the theodolite; left or right bank set up respectively). The survey set up is completed by programming the ship datalogger (entering the beginning and ending coordinates). This coordinates the system so that navigation and all calculations are computed from the origin (0, 0 point) on the X axis.

With the survey and equipment all set up, the survey is performed by systematically navigating the ship station (boat) over the site and constantly tracking the ship station target prism with the theodolite/laser. Cross sections are run at two meter intervals over the area where trash piles are located, and then changed to fifteen meter intervals over the remaining survey. Random soundings are also taken as needed to complete the survey.

Hydro I comes with its own software package for downloading and processing the data. Several functions can be done, such as editing, computing cuts and fills,

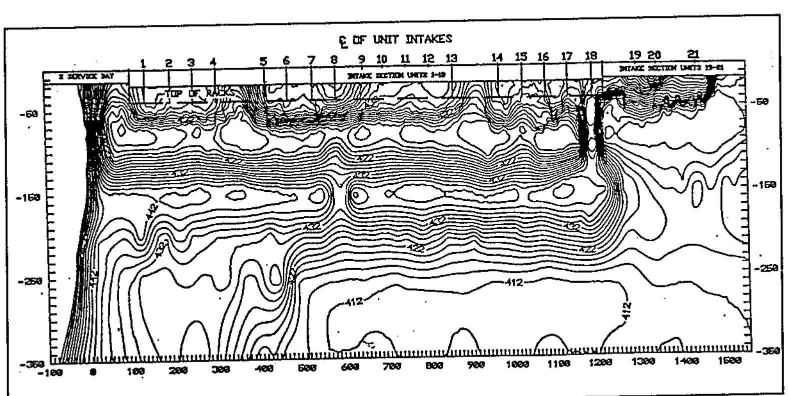


Figure 3: Contour Map of TVA's Wilson Dam

extracting cross sections and profiles from the data base, and also creating printed or plotted reports. In this process the ship datalogger is downloaded and converted to files that can be imported into Hydro I's edit program.

In the editing process, cross sections ran during the field survey can be viewed individually on the computer monitor where data points are checked for distance and depth sounding validity. Hydro I's edit program utilizes a split screen (see Figure 7) where cross sections are displayed on one side and the boat path on the other. The operator is allowed to delete bad distances, correct or delete incorrect soundings, and add extra points to the data base as needed. For example, editing is required when the sonar signal penetrates the trash pile recording the lake bottom and not the top of the trash pile. Editing is very important and one of the more time consuming operations of the process. After editing, these files are combined, and a final file is created where soundings are converted to elevations. This final X, Y, Z file can now be imported to a mapping program or any other program for special processing.

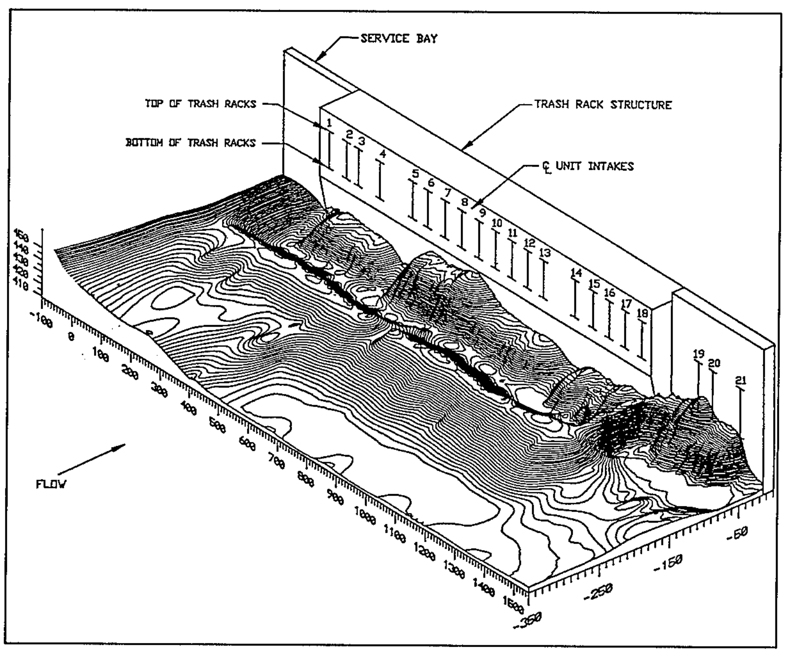


Figure 4: Isometric View of TVA's Wilson Dam

Surfer IV[1] mapping program is utilized for the trash survey mapping process. This program takes the X, Y, Z, data file created by the survey and converts the file to a regularly spaced grid data file. The operator chooses the grid or matrix size; map appearance, accuracy, and contour smoothness is dependent on the grid size (smaller grids yield more accurate maps). Surfer IV uses the grid data file to generate a topographic map, an isometric view, any cross sectional view, and any profile view. During the field survey, a higher concentration of data points are taken over the trash pile areas, and this, coupled with a small grid matrix, yields greater accuracy over these areas. Utilizing this grid file, contour maps, isometric views, cross sections, and profiles can be created using the appropriate Surfer options. Surfer map files can also be imported into a drafting program such as Autocad where structure and other graphics related to the map can be added resulting in a more professional product and greatly improved management tool.

Utilizing the cross section plots, end area information is obtained for the trash at each trashrack. The average end area method is then used to calculate the volume of trash at each rack, and a volume tabulation sheet is then prepared.

After the maps, views, cross sections, profiles, and volume tabulation sheets are prepared, a report is issued.

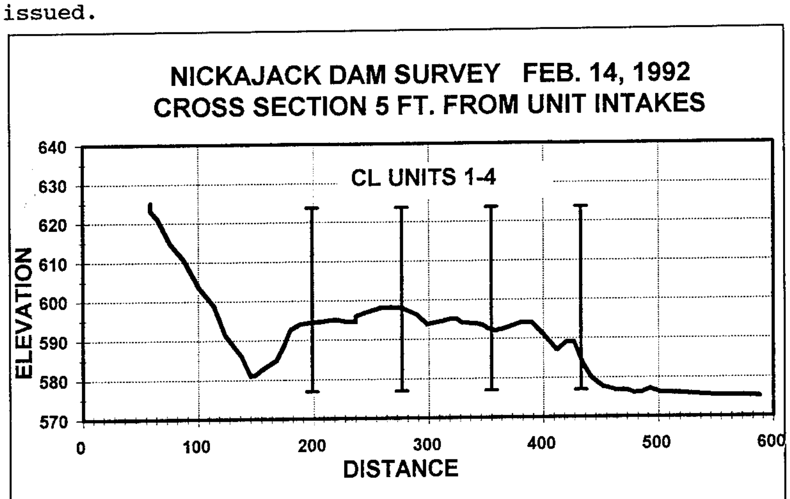


Figure 5: Cross Sectional View at TVA's Nickajack Dam

---

1 - Surfer IV, Golden Software,Inc.,Golden,Colorado.

The report includes the following items:

1 - Site Plan
2 - Survey Data Sheet
3 - Topographic Map
4 - Isometric View
5 - Cross Section Plots
6 - Profile Plots
7 - Trash Volume Tabulation Sheet

When the report for a dam is issued, there are five important results to study and analyze. These are the plan view contour map (Figure 3), the isometric view (Figure 4), the cross sectional view (Figure 5), the profile view (Figure 6), and the volume tabulation sheet. The volume tabulation sheet is usually presented in the report with volumes tabulated for each generator or unit, volume tabulated for distance upstream of the unit, and a total volume is tabulated. The trash volume is used to estimate removal cost and can be used in comparison with future surveys to estimate buildup rates.

The contour map and the isometric view should be reviewed and compared to the bottom elevation of the trashracks to verify any level of trash above the trashrack bottom. A dam forebay with zero trash indicated is shown in Figure 8. The cross sectional view should be reviewed to note the unit that has the highest level of trash accumulation. The profile view should be reviewed to note the maximum trash elevation and the distance that the trash pile extends upstream. The distance the trash pile extends upstream must be worked with the volume tabulations to obtain an accurate estimate of the trash volume.

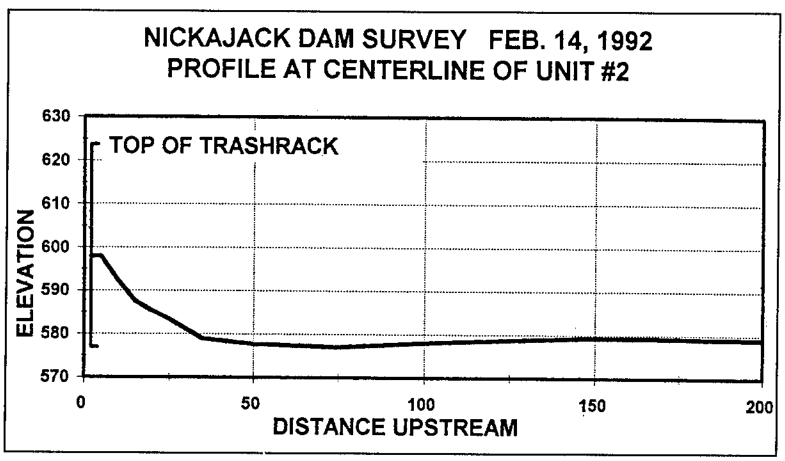


Figure 6: Profile View at TVA's Nickajack Dam

Once all the parameters of the trash are known, engineering calculations can be made to estimate trash induced head loss, load on the trash rack, and decreased power generation. A decision can be made to either remove the trash or leave the trash until future buildup will require removal. Removal can be justified by load on the rack being near the design load, by an economic study that indicates that an improved power generation after removal will compensate for the removal cost, or by trash load and economic study combined. Sometimes the trash must be removed for other reasons such as trashrack repair or painting that has been required by other inspections such as by divers.

The data gathered by the survey should be supplemented by diver inspections of the trashracks to verify the structural integrity of the rack. For instance, a decision may be made based solely on the amount of trash with a load less than design when it was not known that the rack in question was not structurally sound. This could be an unwise decision if the rack were to fail at a load less than design.

Many times it is decided not to remove the accumulated trash pile; however, the data generated by the survey is not wasted data. The survey report should be filed for each dam for comparison to future surveys spanning several years so that a buildup rate can be determined. The rate can be used to estimate when removal will be required for any of the reasons stated above, and the removal can be planned and budgeted along with other work during a maintenance outage.

The surveys of submerged trash that were made at the twenty-six TVA dams provided important data for use in present management of TVA hydro power plants, and the

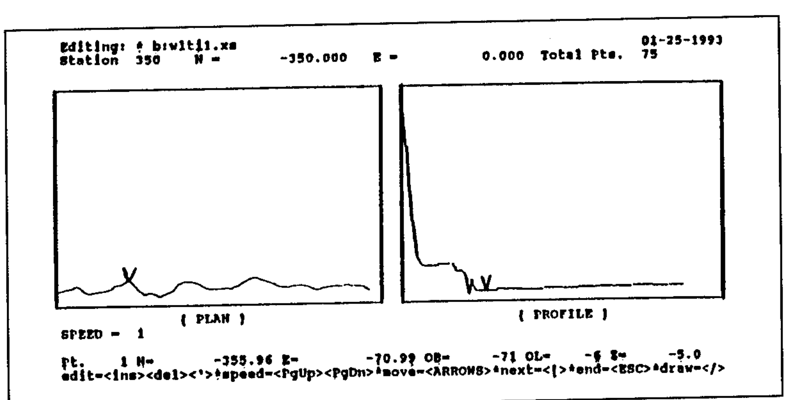


Figure 7: Hydro I Edit Split Screen

data collected will be used in the near future (probably two year intervals) when new trash surveys are performed to compare surveys and estimate buildup rates. The resources required to obtain the surveys were worth expending in return for the benefits. Each survey usually required one day in the field for three personnel. After the field data was taken, one week of work was usually required for one person to edit, process, and validate the data. After the generation of the topographic map, the isometric view, the sectional and profile views, and the volume tabulation sheet, a report was issued which required two personnel two to three days work. This amounts to about fourteen man days of work for a report for each dam surveyed.

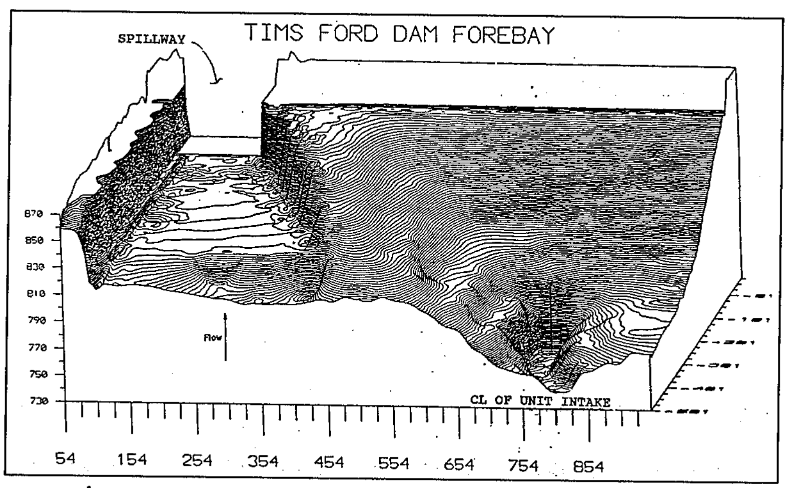


Figure 8: Isometric View of TVA's Tims Ford Dam

SHAFT VIBRATION MONITORING ON VERTICAL UNITS

Rein Husebø[1]

ABSTRACT
Several vertical hydroelectric units have been subject to long-term vibration monitoring, with the objective of minimizing forced outage hours and optimizing maintenance costs. Experience from more than 400 thorough measurements and installation of monitoring systems on more than 40 units have illustrated that simple diagnostic methods based mainly on the analysis of shaft movement, provide good information on the machine condition.

Routines and methods for diagnosis are developed, allowing even non-experts to perform a reliable measurement and diagnosis. The methods are meant to be used by the staff in the power plants, allowing for long term monitoring as well as daily information on the machine condition and behaviour.

Vibration measurements have improved the quality of condition control, by giving information on the condition in a machine while running and without excessive dismantling. In this article, the monitoring systems are described, together with some methods for diagnosis and examples on how to perform analysis.

[1] Senior Mechanical Engineer MSc, Norconsult International A.S., P.o.box 203, 1360 Nesbru, Norway

MONITORING SYSTEMS

A vibration monitoring system is the only surveillance of the mechanical condition in a hydro power unit. Electric and hydraulic properties are most often closely monitored, while temperature monitoring is the only parameter except vibration giving an indication of change in bearing forces. Knowing the bearing forces is of great help in locating the defects and suggesting action.

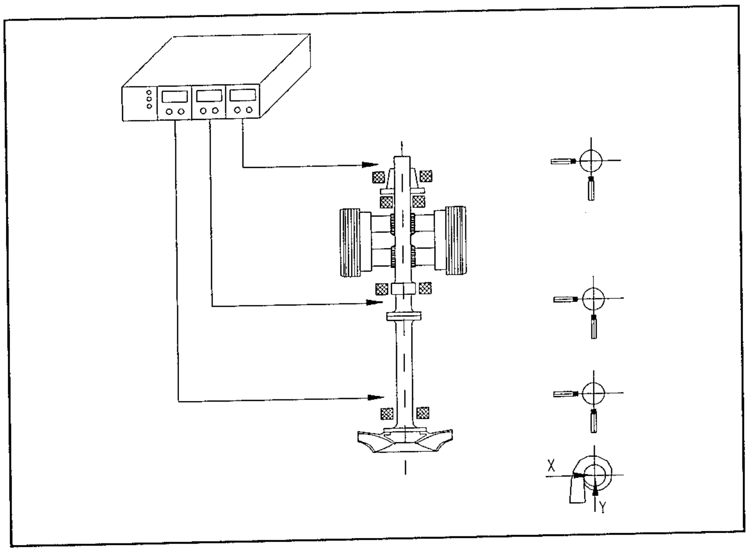
Shaft vibration and shaft position in the guide bearings have proved to be useful parameters when locating the forces acting on turbine, generator, shaft or bearings. In a simple way the size and direction of the forces in different running conditions may be found, hence allowing a detection of changes in forces. Although large and heavy, hydro units are sensitive to changes in forces.

The systems used are based on shaft vibration monitoring, measuring the shaft vibration in two directions at each guide bearing. They are supplied with additional accelerometers or velocity sensors at the turbine, the axial thrust bearing and the generator stator. The monitoring unit is placed in the control room, to simplify periodic readings of vibration amplitudes.

Shaft vibration sensors are placed near the guide bearings. Since shaft vibrations may have nodal points located in the bearings, it is an advantage to place the sensors apart from the bearing center, even at the cost of some accuracy when evaluating bearing forces.

Shaft vibration monitoring will normally detect changes in vibrations at low frequencies, from shaft speed, or lower, up to a few times the rotational speed. Hydro turbines and generators have large masses, and the displacement amplitude at higher frequencies will be small compared to the normally predominant speed frequency. Therefore, accelerometers or velocity sensors should be used to monitor frequencies above 5 - 10 times the speed of the unit, to detect changes at an early stage. (When faults at higher frequency get severe, however, they may be detected at lower frequencies as well.)

On vertical hydro units accelerometers or velocity sensors are used on the turbine, on the generator stator and on the axial thrust bearing. Often the axial thrust bearing is supported on the generator stator, and vibrations from both parts can be measured with one sensor at the bearing.



At stationary conditions in the normal operating range, there is normally small changes in shaft vibration for hydro turbines. Small changes in load, in headwater level and in temperatur have only minor effect on shaft vibrations. The shaft vibrations therefore give reliable longterm readings, and changes in condition may easily be detected, even if running conditions differ between each reading. Changes in tailwater level, however, may have considerable influence on the turbine vibrations.

Monitoring the frequencies below 1000 Hz is sufficient for most hydro turbines with sleeve bearings. For the generator, except for the speed frequency, the frequency of most importance to monitor is 100/120 Hz (passing frequency of the rotor poles). For the turbine, a wider range of frequencies is of importance, mainly the blade passing frequency and several multiples of the speed, the number of runner vanes, guide vanes and stay vanes, and frequencies connected to the oscillating periods in the penstock, draft tube etc. The thrust bearing need monitoring of all the frequencies necessary for the generator and the turbine, as well as the speed times the number of pads.

Proper alarm limit setting is far more important for shaft vibration monitoring systems than for systems monitoring the bearing housing vibration. Most

monitoring systems have two alarm relays for each monitored parameter. As a general rule, it is recommended that one limit is set to give warning when vibrations change from the normal values, and that one is set to give a shut-down when the amplitude exceeds 80 % of the bearing gap. Also, signals should be given when shaft position gets too close to the bearing segments, indicating large pulling forces.

The international standards VDI 2059-5 and ISO-7919-5(draft proposal) cover shaft vibrations in hydraulic machinery. Limits are based on statistical values, and these values may also be a guide when setting proper limits.

Hydro turbines often have an operating range at part load where the turbine shaft vibrations are higher than in the normal operating range. The alarm limits have to cover the total operating range, and even if the part load vibrations may be supressed by use of alarm delay or load dependent limit values, the limits must often be set considerably higher than the vibrations in normal operation. Due consideration must be paid to such effects when setting the alarm limits, to avoid false trips, and to achieve a highly reliable monitoring system.

Monitoring systems most commonly used have alarm relays that will detect faults occuring suddenly in the machine. In hydro units, where most faults develop slowly, systems with manual readings and no alarm relays may be acceptable for many power plant owners, with the same sencors as described above. Systems for this purpose with display but no alarms are also available, at a lower cost than the complete monitoring systems.

DIAGNOSIS METHODS

Vibration measurements in condition control can be divided into four different parts which may be performed separately or as a supplement to each other:

- "Fingerprint" measurements
- Periodic analysis
- On-line monitoring
- Trouble-shooting

In maintenance planning, a "fingerprint" when the machine is in good condition, is a basis for periodic measurements, where results are merely compared to previous measurements. When changes occur in periodic measurements, further analysis may be performed.

A fingerprint consists of the same information as the periodic measurements, but often with additional information, like transients, sound measurements or not-normal running conditions. It may also include a presentation of bearing stiffness and of shaft critical frequencies, properties that are often available from the manufacturer.

Considering the long lifetime of hydro units, with the often slow developing defects, a good basis for maintenance planning will be monthly readings from the display in the monitoring system at stationary running conditions, with annual periodic measurements recording the variations during most of the normal operating range. Registration routines may consist of:

<u>Monthly registrations</u>

- Readings in normal running conditions of:
  - Shaft vibration in each guide bearing.
  - Shaft position in each guide bearing.
  - Axial vibration level in axial thrust bearing.
  - Vibration level in generator stator.
  - Vibration level in turbine bearing or cover.

These readings give information on changes in condition over time. When changes are detected, whole or parts of procedures described in "annual registrations" should be followed to detect the origin of the changes.

Annual registrations
Readings from the monitoring system are taken along with a minimum 7 channel time-record of the shaft vibrations (for units with 3 radial guide bearings), and a time/frequency record of vibrations in stationary parts. The registrations are performed as follows:

Stand-still:	Readings from monitoring system of shaft position in each guide bearing.
Start-up:	Measurement of shaft vibrations in the entire speed range from 0 rpm to nominal speed.
Speed-no-load, unexcited generator:	Measurement of shaft vibrations, vibrations in stationary parts and readings of all values from monitoring system.
Speed-no-load, excited generator:	Same as above.
1/2 load:	Same as above.
Full load	Same as above.
Run-down:	Measurement of shaft vibrations at close to zero speed.

Presenting the vibration parameters as function of speed, load or magnetic force enables a location of most of the forces acting on the shaft system and on the turbine/generator. From the shaft vibrations also the magnitude of the forces can be found when a proper fingerprint analysis has been performed.

For the staff familiar with vertical hydroelectric units, also to non-experts in the field of vibration analysis, locating the forces to component and running condition will be a powerful tool in locating faults and defects, and in confirming unit or component condition. The following examples of registrations illustrate the use of the diagnosis method.

- Speed dependent shaft vibrations is an indication of mechanical unbalanse.

- Change in shaft position during start-up can indikate bearing misalignement. See example next page.

- Change in shaft vibration when exciting generator rotor indicate a magnetic unbalance. The location of the unbalanse on the rotor is given by the key phasor.

- Change in shaft position when exciting generator rotor indicate magnetic pull. This may be caused by uneven airgap or rotor placed excentric in stator.

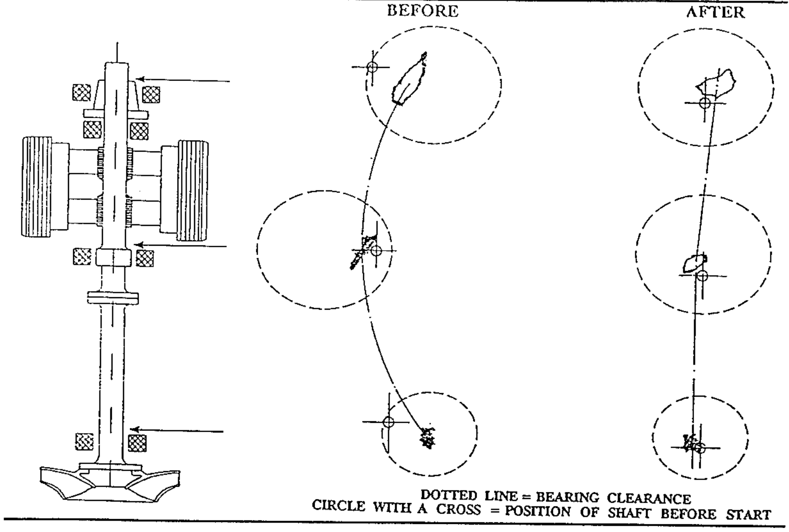
- Change in 120/100 Hz (60/50 Hz line frequency) vibrations in generator when exiting the generator rotor can indicate stator defects.

- Load dependent shaft vibration in turbine indicate a hydraulic unbalance.

- Shaft vibration at very low speed gives information on the shaft alignment.

A system for vibration analysis on hydro power units is available, allowing for the staff at any plant to perform vibration analysis based on easy-to-read presentations. This system is developed for use togehter with monitoring systems, one system covering any number of units.

EXAMPLES OF USE
MISALIGNMENT



A vertical 200 MW Francis turbine with three guide bearings had bearing temperatures above normal values.

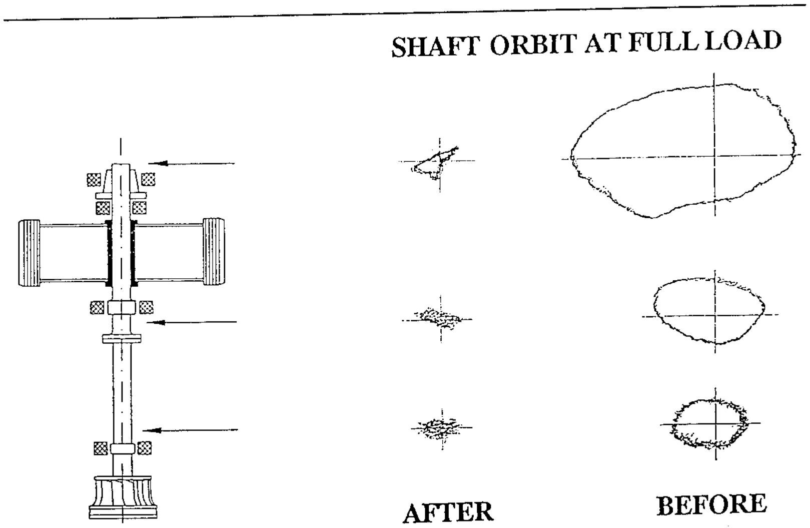
The figure shows the shaft motion in the guide bearings, at idle speed without energized generator.

By comparing the guide bearing gap to the change in shaft position from standstill to normal speed, it could be seen that the shaft position canged in different directions in the lower generator bearing compared with the turbine- and the upper generator bearing. This indicated a misalignment of the guide bearings.

The lower generator bearing had to be moved approximately 0.5 mm.

When looking for changes in foundations, this method is in most cases well applicable.

DYNAMIC BALANCING



A vertical low-head Francis turbine had large shaft vibrations at all three guide bearings, at the rotational speed of the unit. The vibration amplitude was close to the generator guide bearing gap, indicating radial forces of considerable size acting on the bearings.

Shaft vibrations increasing with speed indicated a mechanical unbalance. A decrease in vibration when exciting the generator indicated a magnetic imbalance in a different direction from the mechanical imbalance.

Balancing of the rotor was recommended. Based on one measurement with the non-contacting sensors in the monitoring sys-tem, necessary balancing weights were calculated and location stated. 80 kg of weights were placed at a given position, as a compromise between mechanical and magnetic balancing.

At runaway speed the magnetic forces are not present. When balancing a generator to compensate for magnetic forces, the vibrations at runaway speed must therefore be considered.

VIBRATION FREQUENCY SPECTRUM ANALYSIS

C.F. MALM P.E. - IEEE Member[1]
and Jim Wallace[2]

ABSTRACT

In 1985, our client installed an 800 KW, 360 RPM Pelton turbine with speed increaser and 1200 RPM generator. The Pelton runner was mounted on the extended shaft of the speed increaser. There were no turbine bearings; all of the over-hung runner load and nozzle forces were carried on the speed increaser bearings. Within two years of operation excessive gear wear, and bearing failures occurred. Over the next three years the client tried various "fixes" to no avail.

A vibration frequency spectrum analysis revealed misalignment between the turbine and speed increaser, significant vibration in the axial direction, and the speed increaser case resonated very near the gear mesh frequency.

VIBRATION

Of all the parameters which can be measured on a piece of moving equipment, the one containing the most information is the vibration signature.

Vibration is the oscillation of a body about a reference point. For instance: the shaft of a generator vibrates relative to the bearing, and the bearing vibrates relative to the bearing housing.

---

[1]Principal, C F Malm Engineers, 1100 Virginia Street, Suite 215, Seattle, Washington 98101

[2]Manager, Predictive Maintenance, Eastern Electric Apparatus Repair Company, Inc., 101 Business Park Drive, Skillman, New Jersey, 08558

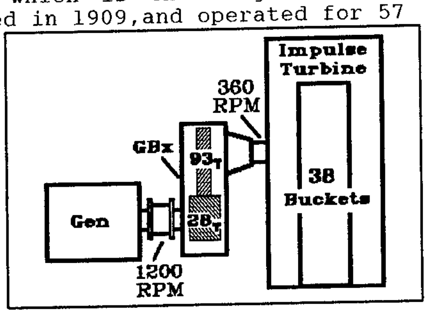
Vibration occurs when some force (forcing function), causes the system to respond to that force. The amplitude of the vibration depends on the forcing function, the mass and stiffness of the system, and the amount of damping in the system.

The vibration signature of a machine is the plot of vibration amplitude vs. frequency.

From the vibration signature we can identify a number of problems such as imbalance, bent shaft, eccentric rotor, misalignment, looseness, broken rotor bars, bearing problems, and runner rubbing on wear rings. Multiple faults are often the case when one problem has been allowed to persist, and it in turn causes additional problems.

HISTORY

The hydroelectric plant which is the subject of our discussion was constructed in 1909,and operated for 57 years with a capacity of 300 kW. The 1909 Hydro Plant was listed as an historic landmark. The historic landmark designation established an important constraint which would affect the rehabilitation of the abandoned Hydro Plant. The powerhouse's visual integrity had to be preserved.



Frequencies of Interest

Turbine . . . . .360 RPM (6 Hz)

Generator . . . . .1200 RPM (20 Hz)

Gear Mesh . . . . 33,600 RPM (560 Hz)

Runner Bucket Frequency . . . . .13,680 RPM (228 Hz)

Difference in input/output speed . . 840 RPM (14 Hz)

Bearings: SKF 23948C, TMK 219122/EE219068,
TMK 93125/93825

Figure 1

In 1983, the owner, a town of 15,000, voted to issue revenue bonds to restore, and upgrade the historic hydroelectric plant.

The Owner's consultant found that the 24-inch, 4,700 foot-long riveted penstock, which was built in 1924, could continue to service the water treatment plant and hydroelectric plant. The consultant's plans called for an impulse turbine, operating at 350 feet of head, at 360 RPM. Constrained by the historic powerhouse, the larger diameter of a 360 RPM generator was not possible, and a speed increaser was specified.

Turnkey proposals were solicited in 1983 and the successful bidder was a local electrical contractor with some hydroelectric experience.

The contractor decided to assemble the system himself and placed separate purchase orders for turbine, speed increaser, generator, switchgear and control system.

The horizontal turbine was an 850 kW, two nozzle, 19 bucket impulse machine operating at 360 RPM. The turbine was an "over-hung" design with no bearings. The runner was to be mounted on an extended shaft of the speed increaser. The speed increaser selected was a 1:3.33 step up to 1200 RPM. The generator was a 1200 RPM brushless synchronous machine.

The plant was put on line in 1985. At startup, the long speed increaser input shaft was actually whipping, and the speed increaser manufacturer installed an extended bearing housing to help support the over-hung load. The speed increaser manufacturer had not been advised that the runner was to be an over-hung load on the speed increaser input shaft. Since the problems were manifested in the speed increaser gear box as bearing failure and gear wear the speed increaser was, of course, the suspected offender. The problems appeared to be in the speed increaser because in addition to transmitting the torque from the turbine to generator, the speed increaser was also carrying the weight of the turbine runner and nozzle forces.

During the six years of service until 1991, there had been several bearing failures and the gears were turned around to use the unworn side. Figure 1 is a schematic sketch of the system.

VIBRATION ANALYSIS

Vibration analysis of the Pelton turbine-generator showed no signs of looseness, bearing problems, or electrical problems. However, this unit did show problems in the gears of the speed increaser.

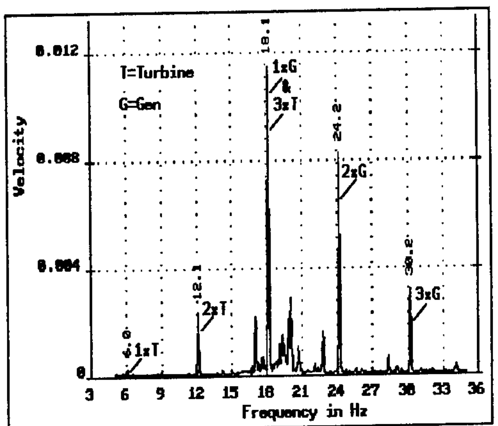


Figure 2

The input shaft from the turbine showed signs of misalignment. Note the higher second harmonic of the turbine input shaft in Figure 2.

An impact test on the front face of the gear box, just under the case split, between the input and output shafts, showed a resonant area that is very close to the gear mesh frequency of 560 Hz (see Figure 3). The rule of thumb for proper design requires that no fundamental component frequency be within ±20% of any major running frequency, in this case the gear mesh frequency. This resonance greatly amplified any vibration caused by the meshing of the gears. The presence of resonance at the gear mesh frequency causes any vibration source to be multiplied many times. The misalignment noted in the gears could easily be amplified by the resonance to levels which could

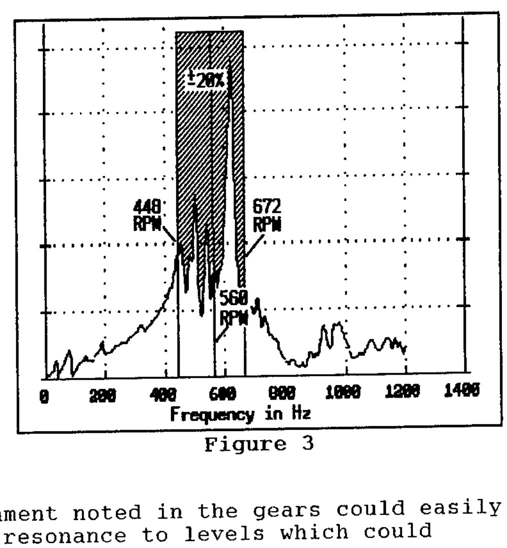


Figure 3

severely shorten the life of bearings, generator windings, and the gears themselves.

It was noted that the gear mesh vibration went up by a factor of 4 when the generator was placed under load. This is consistent with the resonance findings, and could also be an indication of an eccentric gear condition.

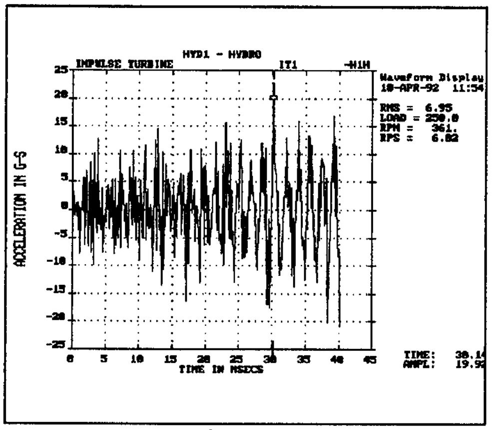


Figure 4

The time waveform measured in G's clearly showed impacting at gear mesh frequency. The waveform showed some amplitude modulation which is common when a gear runs within resonance.

Some of the larger impacts were measured from about +19 G's to -14 G's. This equaled total G force impact of over 30 G's. This amount of impact energy will probably cause teeth to be broken from the gears within a few years or less. Experience tells us that gears of this size and speed should not incur more than 5 or 6 G's impact forces, and it is common to see less than 2 G's impact force.

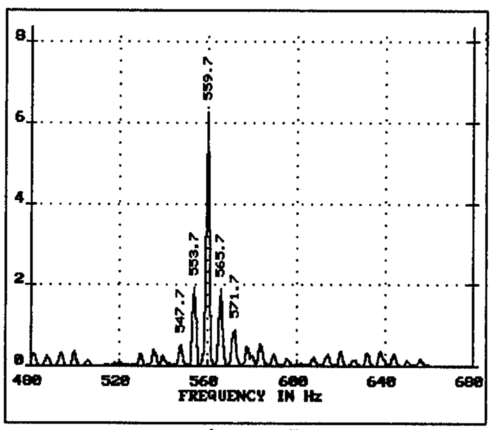


Figure 5

The spectra of the gear mesh frequency (GMF) showed sidebands modulated by the running speed of the input gear. This indicates that the source of the vibration is the bull gear. A visual inspection of the gears showed

the bull gear. A visual inspection of the gears showed offset wear patterns on the front face of the bull gear teeth, and confirmed the bull gear as the source of the vibration. The pinion gear also showed excessive varnish plating from running too hot in the type of oil used.

The over-hung pedestal bearing on the input shaft from the turbine appears incapable of securing and maintaining the alignment on the input shaft and is probably adding to the overall problem.

The stator case on the generator also showed a resonance near the running speed of the generator (Figure 5). This too added to the overall vibration on the gears and the generator. There may be a resonant soft foot (i.e. a tapered or twisted foot) in this machine. This would cause a twisting force on the generator case which could bring the resonance near the running speed.

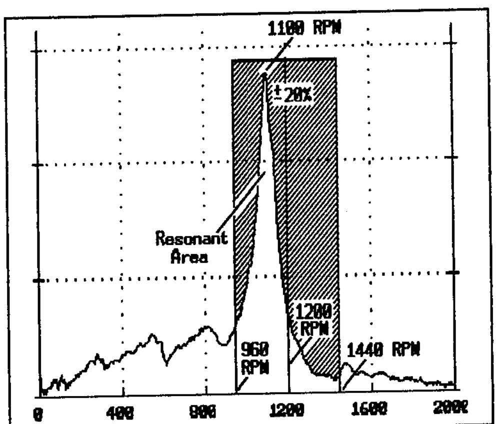


Figure 6

Even though the alignment had been checked several times, the machine was re-aligned by a skilled millwright. He found the runner was not in the same plane as the nozzle and the nozzle jets were approximately .040 inch off from the bucket centers. It appeared that the misalignment had existed from the initial installation, and that turbine, speed increaser and generator had not been installed such to make future alignment adjustments possible. With the runner over-hung on the speed increaser shaft, moving the speed increaser was the only possible way to align the runner to the nozzles. Moving the speed increaser then caused misalignment between it and the generator.

FINDINGS

The correction of the input shaft misalignment successfully corrected the excess energy at gear mesh frequency.

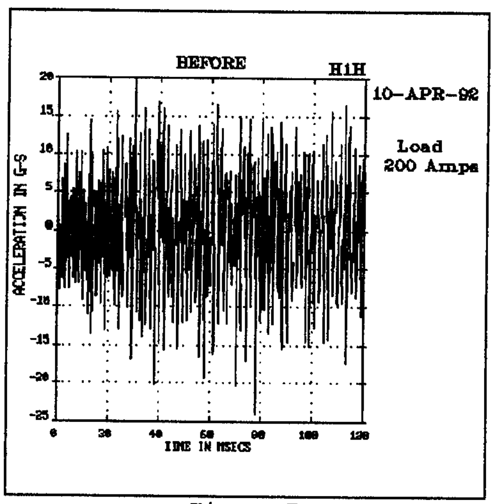


Figure 7

Prior to re-alignment, the gears were showing impacts as high as 20 G (Figure 7). Following alignment the time waveform sample at the same position, same load, showed impacts are now at and below 6 G's (Figure 8). This level of energy is more consistant with the size and speed of this unit.

The alignment of the turbine shaft has lowered the gear mesh energy to 1/5th of its previous amplitude. This has effectively reduced the forcing function that was driving the gear case resonance shown in Figure 3.

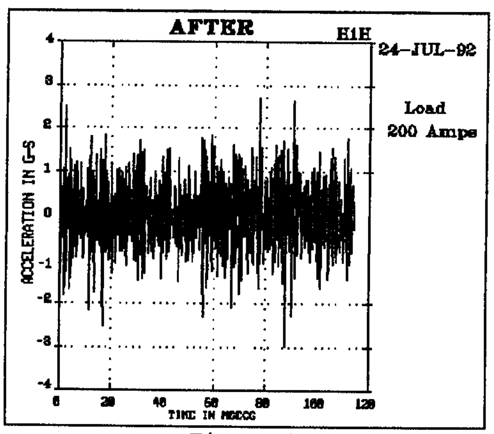


Figure 8

The vibration spectra of the running speed components and their harmonics show that the alignment between the generator and the gear box became worse (Figures 9 & 10). This vibration level will shorten the life of the gear box output shaft bearings and the generator bearings.

The overall vibration amplitude at the vertical position on the inboard generator is .31 ips. in the current spectra as compared to 0.044 ips in the prior spectra.

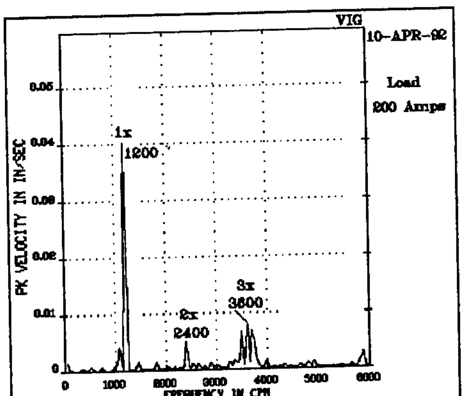


Figure 9

The source of the 120 RPM sidebands around the 3600 RPM third generator harmonic was not positively identified. The bearings frequencies did not fit. This frequency seems modulated by the turning speed of the input and output gear. Apparently, there is a rough area on the input gear, and a similar rough area on the output gear, and these are coming in contact about 120 times every minute. This fit the pattern. The energy is low level and if it is not from a bearing its effect should be negligible.

CONCLUSION

The Owner wants a "once and for all-time" fix. A turbine bearing is being designed to take the load off the speed increaser. The speed increaser is being replaced, and the support structures are being modified to facilitate re-alignment.

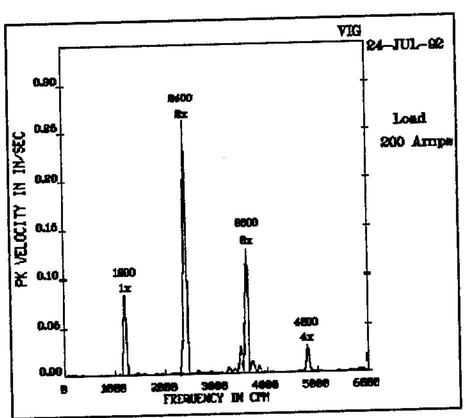


Figure 10

It appears that design of an over-hung runner on such a short moment arm as provided by the speed increaser bearings is at best, marginal. A proper alignment initially would have minimized the wear on the speed increaser to the point it most likely would still be

serviceable. A vibration analysis would have cost only 1/100 of the cost of the repairs, not including the cost of repairs and downtime for the past six years.

If a vibration signature is obtained at the time of start up to establish a baseline, and subsequent signatures are periodically compared, incipient problems such as bearing failures, anchor bolt looseness, and misalignment, can be corrected before they cause other problems and become a catastrophe.

The paper that was to have appeared on pages 1804–1811 has been withdrawn by the author.

CONSIDERATIONS FOR DEWATERING HYDRO UNITS AND GATES

Richard M. Rudolph[1], John R. Kries[2]

INTRODUCTION

Northern States Power Company (NSP) owns and operates hydro power facilities dating from the turn of the century to the 1980s. Dewatering mechanisms at these facilities range from timber stoplogs to steel head gates to nothing. NSP's goal is to use the safest, fastest, and most economical method of dewatering, short of reservoir drawdowns. Project experiences on dewatering schemes NSP has considered or used are presented in this paper.

AGE AND TYPE OF EXISTING FACILITIES

The type and age of dewatering mechanisms for hydro components at normal pool varies. Most of NSP's facilities are old and some facilities were purchased from others. Therefore, structure plans are generally incomplete and, in some cases, are not available. Drawings have not always been prepared, further complicating dewatering schemes. For example, stoplog slots have been partly filled with concrete with no records available.

The types of dewatering schemes designed for existing facilities are:

1. Radial gates (multiple arm with turnbuckles)
2. Stoplogs
3. Steel flap gates
4. Roller and gear mechanism gates (wood and steel)
5. Steel valves
6. Steel frames with wood needles

[1]Richard M. Rudolph, Administrator, Hydro Engineering, Northern States Power Company, 100 North Barstow Street, Eau Claire, Wisconsin 54702

[2]John R. Kries, Supervisor of Hydro Maintenance, Northern States Power Company, 100 North Barstow Street, Eau Claire, Wisconsin 54702

Bottom sills composed of wood, concrete, or steel vary in shape and condition. The steel components are often riveted back to back channels, which can inhibit sealing. Concrete varies in quality and condition and is generally non-air entrained.

CONDITIONS OF DEWATERING MECHANISMS

Some existing dewatering mechanisms were not properly maintained. As the need to inspect and repair turbines and other structures increased, the reliability and condition of the dewatering mechanisms has become more important. Safety of men working in the area is of particular concern. Mechanisms, such as gates, are in wet environments. The condition of submerged bearing surfaces for mechanisms are always questionable. Deteriorated bearing surfaces encountered below normal pool are often:

1. Deteriorated concrete in the regions from the surface to about 2 feet below the freeze/thaw line and near drawdown limits.

2. Deeply pitted steel members where air entrainment occurs, such as in high velocity zones.

Submerged gates are difficult to maintain and typically exhibit excessive deterioration. These gates often have sealed chambers in all or part of the gate to provide buoyancy for lifting. However, they may not be functional. Evaluating whether the gates should be rehabilitated, replaced or abandoned is often difficult.

ALTERNATIVES -- PLANNING

Determining the appropriate type of dewatering scheme requires:

1. Reviewing existing plans and contacting present and former employees who worked on project;
2. Verifying dimensions, elevations, and plumbness;
3. Defining seasonal conditions that affect project, such as streamflows, generation, and weather;
4. Inspecting sealing surfaces;
5. Determining condition of components that might be used for structural support; and
6. Selecting experienced construction personnel.

Upon completion of data research and inspections, operations, engineering, and construction personnel meet to evaluate dewatering options. The issues addressed include:

1. Worker safety.
2. Access constraints for equipment and workers.
3. Weights and sizes of material to be lifted by hoists or cranes. (Hoists outside of powerhouses are generally inadequate for expected lifts.)
4. Weather and river flows.
5. Cost of lost generation.
6. Whether the loss of flow capacity (i.e., gates) will affect safety of dam. Winter weather (with lower flows) versus warm weather (with potentially higher flows) must be considered.
7. Drawdown limitations that vary by location and season.
8. Environmental impacts.

9. Time required to install and complete work, including design, procurement, and fabrication.
10. Types of dewatering schemes available.
11. Whether scheme plan can be used at other facilities or is available at other facilities.
12. Whether dewatering schemes should be permanent. What is the frequency of dewatering and will it serve additional purposes? For example, a unit may be used to shut off flow for run away turbine/generator.
13. Cost for one-time versus multiple usages.
14. Equipment needs and personnel experiences suited to this project.

DEWATERING SCHEMES

The types of dewatering schemes currently used on NSP facilities are: (1) hinged floatable bulkheads; (2) steel head gates or bulkheads; (3) wood and steel stoplogs; and (4) steel frame with needles.

Floatable Bulkheads

Floatable bulkheads were first developed and used at the Wissota Hydro Project. This concept has been subsequently used by others. The bulkheads shown on Figure 1 are steel caissons hinged together to act like a garage door across the opening. Buoyancy of the units controls the rate of descent and pressure on the bottom seals. The units are designed for depths of 30 feet and are about 36 feet long by 3 feet 9 inches high per section. The first usage (Wissota) included guides on the piers to control lateral movement during installation. Recent changes in installation have eliminated use of guides.

Steel Head Gates

Steel head gates contain horizontal beams attached to side members (i.e., channels) with an upstream steel skin plate welded to them. The bottom member bears on the sill or connecting surface of multiple units. Allowable weight and height per unit depends on equipment handling capabilities. Side slide members are used to facilitate installation. Rollers are optional and filler valve size matches volume being dewatered. For sealing, various items from steel to steel rubber flaps are used. Figure 2 is a typical head gate.

Stoplogs

Stoplogs can be timber, timber lagging with steel beams, or steel alone. Stoplog design is based on depths of water and span between support points. The thickness of wood is a function of wood type and properties. The stoplog slot concrete and steel conditions vary. Continuous bearing surfaces from the stoplogs have been enhanced by securing several stoplogs together by using threaded steel rods.

Steel stoplog weights can be a disadvantage due to the need for additional hoisting capacity. However, cost of wood currently is high and steel components are often more economical for longer spans or deeper depths. Figure 3 is a typical steel stoplog. The steel stoplogs approach head gate concept. The main control for sizes and weight of each log is the lifting mechanism.

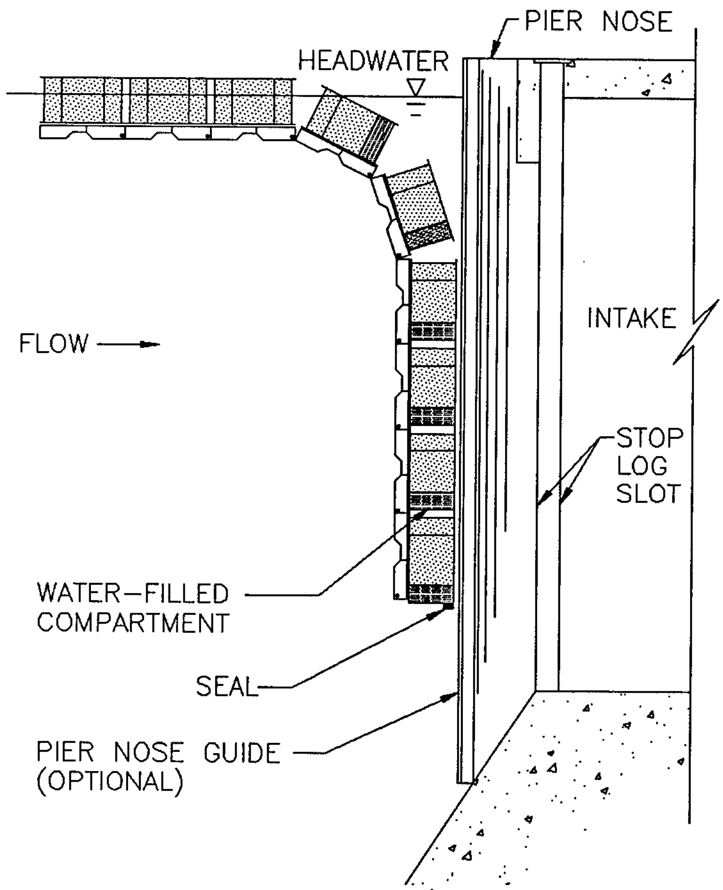


FIGURE 1
FLOATING BULKHEAD

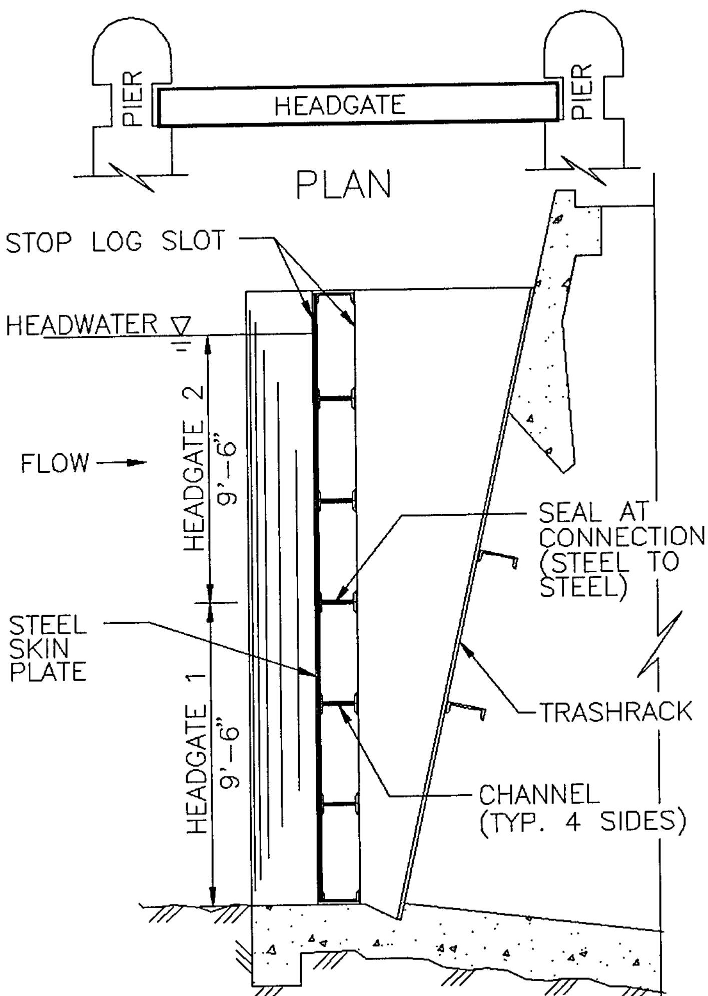


FIGURE 2
HEADGATE

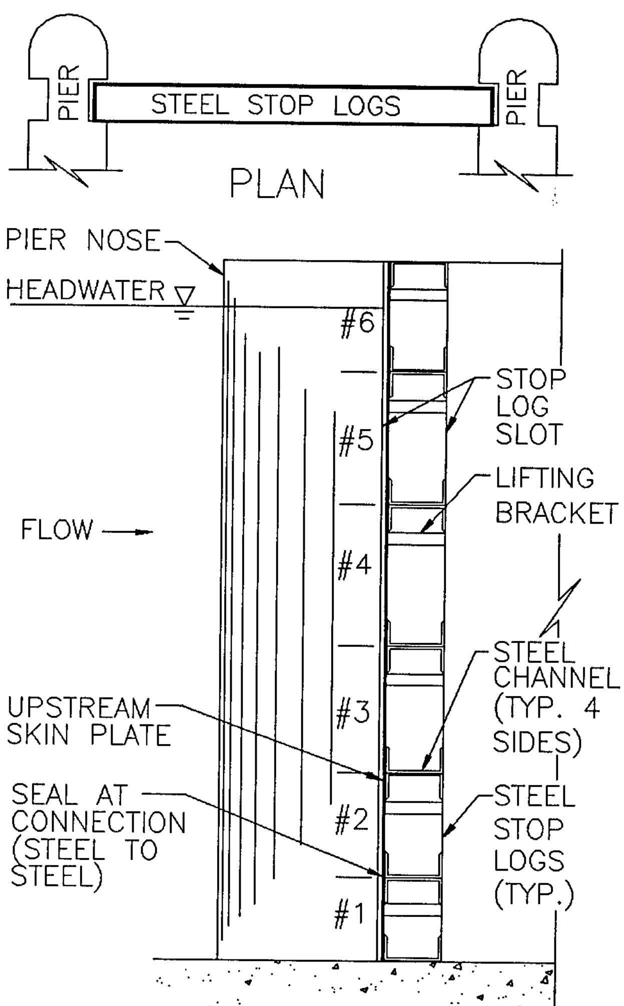


FIGURE 3
STEEL STOP LOGS

Steel Frame with "Needles"

This system consists of horizontal steel members that act as guides and support beams for vertical wood or steel members (needles). Generally, needles of interlocking steel sheets are economical and provide adequate sealing. This system is generally most economical for shallow depths and longer spans. Figure 4 is an example of a "needle" cofferdam.

Which type of dewatering scheme is best? All aspects of the work should be considered, including time, cost and safety. The projects described below illustrate typical use of these alternatives.

PROJECT 1

This project required dewatering of a hydro plant for turbine repairs. The work included removal of horizontal turbines. The 1910 vintage intermediate stoplog columns are riveted steel construction.

Underwater inspections revealed that steel was deteriorated with a reduced load carrying capacity; thus safety was a concern. If the frame even partially yielded, which was the projected failure mode, water would flood the powerhouse. With the turbines removed, direct water inflow to the powerhouse operating offices and generators would occur with a failure. A bulkhead could be installed over turbine shaft opening once the unit was removed to reduce the access for flood water, but concern for failure remained. Monitoring the steel frame for deflection was considered, but was rejected for safety reasons.

Concerns for the environment and potential impact on the adjacent paper industry also weighed into the decision.

The options were to leave the unit out of service or formulate a plan to dewater with full pond.

The inspections, plant economics, and review of the structural plans revealed the following:

1. Vertical columns were part of the truss arrangement that supported trash racks.
2. Dewatering schemes had to be applicable to two adjacent units.
3. Costs had to be minimized since the project's estimated economic remaining life is about 10 years. Generation pay back needed to be shorter than this.
4. Work could be done in the winter or summer months.
5. Access is limited for mobile cranes as well as the existing lifting capacity.
6. Dewatering scheme needed to use upstream noses of the piers for a sealing surface.
7. The bottom intake slab is parallel to the pier noses, which created potential uplift and sealing concerns.

Based on the preceding, two options were considered: (1) use of floatable bulkheads, and (2) installation of a frame on the piers for stoplogs or "needles." Based on the depth of water and previous experience, the floatable bulkheads were selected.

The floatable bulkheads required modification of the bottom seal and other miscellaneous steel changes. The units were unloaded upstream and floated down to the plant. Once assembled at the site, the units were installed within several hours, after some obstructions were

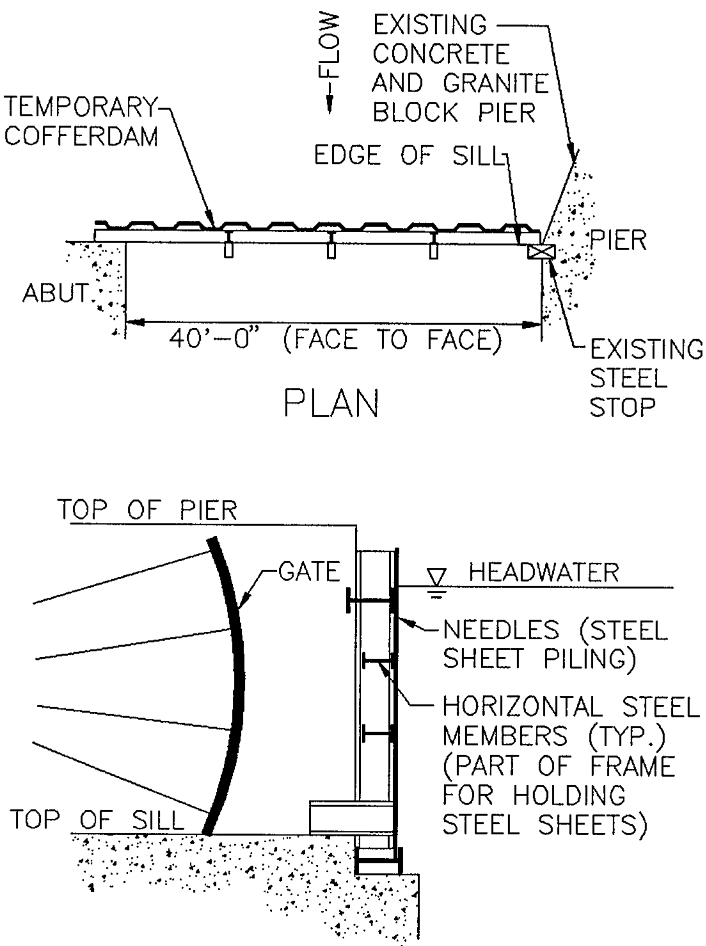


FIGURE 4
NEEDLE COFFERDAM

removed. Removal times for bulkheads was approximately one hour. Before being used in an adjacent unit, the bulkheads were secured like a floatable dock. The floatable bulkhead concept was again used successfully. Currently, the units are stored awaiting another application. They have been paid for many times over.

PROJECT 2

This project required dewatering a tainter gate, which is 40 feet wide and 17 feet high, for repairs. The gate and adjacent fixed crest spillway is used for discharging flood flows. Figure 4 shows the project section with the needle arrangement.

Based on underwater inspection, review of old photographs, discussions with existing and retired operators, and review of available plans, the following was learned:

1. No stoplog slots were available. Original dewatering method was for needles. A one-piece floatable unit could be used.
2. The site hoist equipment was removed.
3. To minimize need of gate for floods and loss of flashboards, the best time to do work was between November 1 and March 1.
4. Crane usage would be limited, given distance from dam centerline to staging pad area.
5. Drawdown was not practical since the gate is a primary source for passing excess flows.
6. Anchorage on piers was limited and the sill condition was questionable.

Therefore, two alternatives were considered: (1) floatable bulkheads, with two units secured together to meet horizontal spacing requirements; and (2) frame and needle section. Bulkheads from a previous project were available; however, the cost of transportation and required modifications would have been more costly than a new frame and needles. Once the decision was made to use a frame and needles, the projects constraints were time and weather.

Time components for the entire project included: (1) design of cofferdam, (2) material ordering and fabrication of frame, (3) installation (since a mobile crane was needed), (4) repairs to gate, and (5) removal of cofferdam. The cold weather and threat of early spring runoff were concerns for items 3 through 5.

Due to the weights of cofferdam components, in addition to time and weather constraints, the decision was made to dissemble and transport the tainter gate to shore. The mobile crane used for installing the cofferdam could be used to off-load the gate providing dual usage. Once on shore, repairs could be made concurrently with sill, side plate, and related concrete repairs, reducing construction time.

Fast track design and prompt engineering decisions were required. Despite the ongoing communications required of all involved, the project was completed on time and in a safe manner. Planning was important, but rapid response during construction was even more important. Also, a highly talented crew that developed ideas as well as completed work was essential for a successful project.

PROJECT 3

This project included dewatering two adjacent hydro units to inspect flap gates and repair turbines.

Based on previous difficulties in dewatering units and underwater inspection, the following was noted:

1. Upstream stoplog slots in concrete piers were deteriorated down to 3 feet below water surface.
2. Dewatering one unit alone was very difficult, due to a connection between the units.
3. Reservoir drawdown was not considered feasible due to flow requirements and environmental concerns.
4. Condition of the flap gates was questionable. Repairs were difficult to ascertain given the number of members including sealed sections.
5. No on-site cranes were available, so lifting was limited to mobile units.

Based on the preceding, stoplogs or head gates were considered the most economical options. The existing wood stoplogs were used for tailrace dewatering. Steel head gates, shown on Figure 2, were built. The section sizes were primarily based on the capacity of the mobile crane to be used. By having large bulkhead sections, the concrete bearing surfaces that were eroded near the surface could be bridged. Sealing between units was simply steel on steel. The simple design was very economical to fabricate.

The inspection of flap gates revealed severe deterioration. The decision was made to remove them and construct new stoplog slots downstream of the bulkhead. Future work is to construct monorail and storage racks for head gates that would be installed in the new slots. The two temporary head gates will be modified so they can be reused with little additional cost. The other two bulkheads will be used, with minor modifications at another site for tailrace dewatering, replacing current wood stoplogs.

SUMMARY

Each dewatering project has unique problems. The most appropriate scheme to be used for dewatering requires planning with all parties involved. NSP involves construction crews, operations, and engineers to evaluate site conditions, construction restraints, and design limitations to have more complete planning. This interaction promotes team involvement, which results in safe, economic, and timely projects.

In reviewing the older design concepts, one sometimes finds that old ways (sometimes with updated materials) are still the best ways. In other cases, new ideas are paving the way, because each hydro facility has its own set of problems.

PROTECTIVE COATINGS TO RESIST CAVITATION EROSION IN CONCRETE HYDRAULIC STRUCTURES

Joe Wong[1]
Dick Brighton[2]

Abstract

This paper describes the use of polymer coatings to protect concrete hydraulic structures from cavitation erosion. A laboratory test program was performed to identify materials which are resistant to cavitation erosion. These materials were then tested in the field. The coatings were applied to sluiceways where cavitation damage had been observed.

Introduction

Cavitation damage to concrete hydraulic structures is a recurring problem for electrical utilities, particularly in structures where both the velocity and vorticity of the hydraulic flow are high. Ideally, cavitation in the hydraulic structure should be eliminated in the design phase. While this objective may be theoretically possible, in practice it may prove to be too costly or impractical to implement. In some instances steel plates has been installed to minimize the cavitation erosion. In other instances to improve the cavitation resistance of the concrete, fibre reinforcement, polymer impregnation, aeration, and even the selection of the type of coarse aggregate and sand grain shape have all been considered (Schrader, 1987). Despite these modifications, cavitation erosion still occurs under certain high velocity flow conditions. Since cavitation can only be controlled and not completely eliminated, regular maintenance is a normal part of operating expenses. Unfortunately, repairs involving even the use of high quality concrete with a good surface finish provide little improvement in resistance to cavitation. When properly installed, steel plates will resist cavitation erosion. However,

[1] Materials Engineer, Powertech Labs Inc., 12388 88th Avenue, Surrey, B.C., Canada, V3W 7R7

[2] Civil Inspection Engineer, BC Hydro, 6911 Southpoint Drive, Burnaby, B.C., Canada, V3N 4X8

they are expensive to install and it is difficult to control the erosion at the edge of the steel plates. Consequently, there is a need to develop suitable cavitation resistant lining materials with good adhesion to concrete substrate and ease of application.

B.C. Hydro and the Canadian Electrical Association have jointly funded a research program to minimize or alleviate cavitation erosion in concrete hydraulic structures. The main focus of the program was to identify and develop polymeric protective coating systems which would resist cavitation erosion. The coatings were then applied to a concrete hydraulic structure for field testing.

Laboratory Testing

The success of any repair system must satisfy a number of conditions:

- resist cavitation
- exhibit good bond strength to the concrete
- resist exposure to sunlight
- resist exposure to extreme temperatures
- resist impact damage
- be easy to apply

Three classes of materials were considered for testing:

• Polymeric Coatings	- resins applied to the concrete surfaces
• Reinforced Polymers	- resins reinforced with fibres or powders.
	- glass, steel, or thermoplastic fibres
• Lining Systems	- tiles adhesively bonded to the concrete
	- metallic or thermoplastic tiles

Screening tests were performed in the laboratory using a cavitation jet testing apparatus called Cavijet (Cheng, 1987). The jet is specially designed to create cavitation bubbles downstream of the jet nozzle which implodes onto the test sample.

The samples which showed the best cavitation resistance were then subjected to physical testing to ensure that the materials are compatible to the concrete substrate and are durable when subjected to field conditions.

The cavitation tests showed that linings systems with thermoplastic tiles show the best resistance to cavitation. A tiling system was devised which bonded very well to the concrete. However, thermal compatibility tests showed that the thermoplastic liner exerted excessive tensile stresses to the base concrete under extreme temperatures resulting in failure.

Based on the testing the following materials were chosen for field testing:

Coating A - a modified epoxy resin with silica filler.

Coating B - an epoxy resin with a urethane powder filler.
Coating C - a two component polymer
Coating D - a filled epoxy reinforced with thermoplastic fibres.

Field Testing

A test site was chosen at the Hugh Keenleyside Dam, an earthfill and concrete structure located on the Columbia River, 8 km upstream from Castlegar, B.C. The 50 metre high concrete structure shown in figure 1 controls a 3,650,000 hectare drainage area and holds back a storage reservoir extending 232 km north to Revelstoke. Release of the 11.7 billion cubic metres of live storage is controlled by four 15 metre wide sluiceway bays and eight 6x7 m low level ports.

Since its commissioning in 1967, Hugh Keenleyside dam's concrete sluiceways have been susceptible to cavitation erosion. Figure 2 shows the damage to the concrete sluiceway. Damage to the sluiceway invert has occurred around the gate slots for the operating gate in all four sluiceway bays. As the sluiceways are regularly used for 6 to 8 months of the year, this damage is a continuing problem and has been repaired at least three times previously.

A study conducted by B.C. Hydro's Hydrotechnical Department (Cass, 1987) (Banks, 1990) concluded that the severe cavitation at and around the foot of the gate slots is a result of cavitation from vortices generated in the upstream corners of the gate slots during small gate openings. The study also showed that the sluiceway crest is designed against the occurrence of negative pressures. Therefore the cavitation does not occur across the width of the sluiceway bay, but is confined to small areas around the gate slots. These small areas of cavitation damage are ideal test sites for evaluating the performance of cavitation resistant coatings.

Sluiceway Repairs

Sawcuts were made in the concrete at least 150 mm beyond the cavitated areas. Cuts were made at least 50 mm deep and were slightly angled (about 5 degrees) in order to achieve a "keying action" with the concrete repair. Concrete within the cavitated area was chipped away to sound concrete. Loose material and laitance was removed from the repair area using a waterjet. The concrete surface was kept saturated until the repair material was placed. The approximate area of each repair was 4.7 m^2 (50 ft^2.).

Since the cavitation damage was as deep as 30 cm, a cementitious backfill material was required. A prebagged fast set concrete backfill was mixed onsite and the prepared holes were backfilled. This backfill was allowed to cure for seven days prior to the application of the coatings.

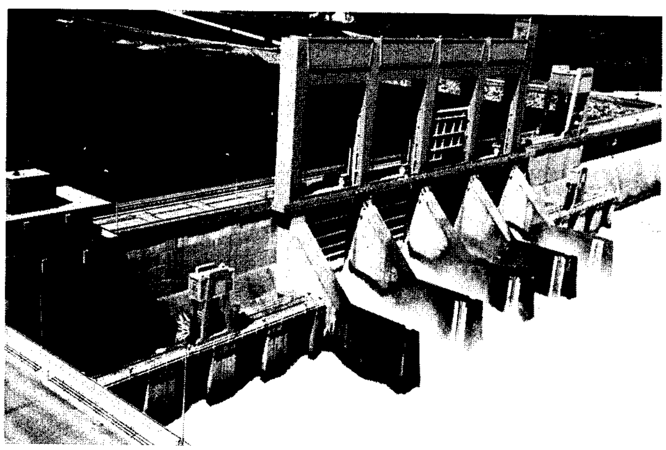


Figure 1: Sluiceways at Hugh Keenleyside Dam.

Figure 2: Concrete erosion due to cavitation. Damage in some areas was as deep as 30 cm.

The application of the cavitation resistant coatings were performed with relative ease. Efforts were made to keep the worksite dry and relatively warm. The materials did not require any special equipment to apply.

OPERATION OF THE SLUICEWAY

In the study by Cass et. al. for B.C. Hydro, it was shown that at the average flow past the gate slots can exceed 12.2 m/sec (40 fps). U.S. Bureau of Reclamation (USBR) recommends that the mean velocity of flow past gate slots be limited to 12.2 m/s (40 fps) to avoid cavitation damage to concrete.

In a second study by Banks et. al. for B.C. Hydro, it was shown using acoustic methods that cavitation in the gate slots can occur with gate openings between 0.6 m to 4.4 m with the highest intensity occurring at 3.0 m. The study indicated that cavitation can occur at reservoir levels above 433 m.

During the first year after the repairs were made, operation of the sluiceways commenced on October 1, 1990 and continued until January 22, 1991. The gate openings ranged from 0.98 to 5.81 m. The discharge ranged from 280 to 2800 m^3/s. The headwater level varied between 430-440 m. Figure 3 shows the gate opening records at Keenleyside for 1990.

During the second year, operation of the sluiceways occurred between June 22, 1991 until February 1992 and the reservoir was operated above 433 m for 200 days. Figure 4 shows the gate opening records for 1991.

SITE INSPECTION

The coatings were inspected after 2 years of exposure to cavitation.

Coating A
Most of the coating was removed from the repair area. Some areas beneath the gate experienced cavitation erosion in the backfill to as deep as 75 mm.

Coating B
There appeared to be minor damage to this coating repair

Coating C
This coating system suffered severe damage in many areas. Erosion to the backfill material was as deep as 150 mm.

Coating D
This coating experienced minor damage.

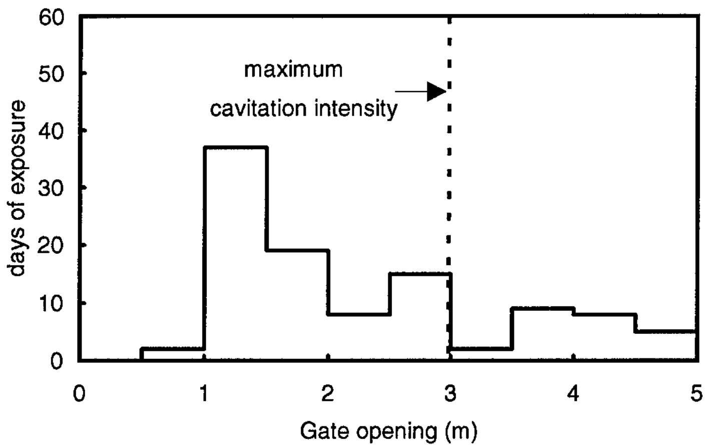


Figure 3: Gate operations, where sluiceways are exposed to cavitation erosion, for 1990.

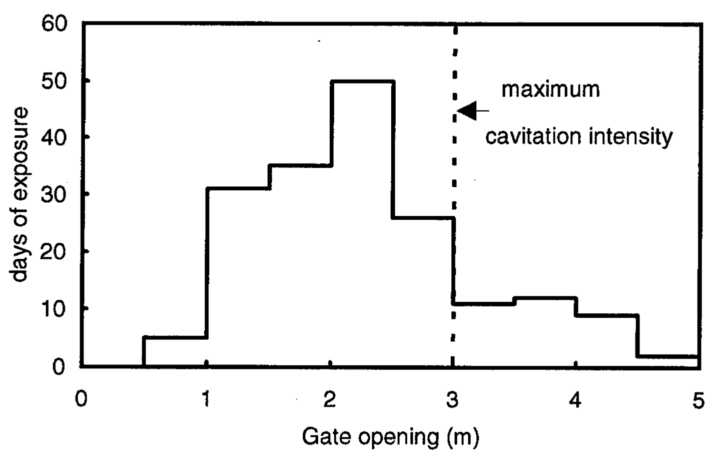


Figure 4: Gate operations, where sluiceways are exposed to cavitation erosion, for 1991.

DISCUSSION

PERFORMANCE OF COATINGS

During 1991, all the coatings were subjected to prolonged cavitation. There were a total of 140 days where the gates were operating at openings where cavitation occurs. There were 30 days where the coatings were exposed to severe cavitation (gate openings between 2.5 to 3.5 m). As a result, all the coating systems suffered some damage.

During 1990, the gates were subjected to only 99 days of cavitation of which 17 days were intense cavitation. It was observed that coatings B & D suffered only minor damage. It has been recommended that B.C. Hydro operate the sluiceway to avoid gate openings that cause severe cavitation. Therefore, it may be possible to avoid any damage to potentially good coatings. It will take a few years to gain enough experience to determine these optimal openings.

It was very important to test these coatings in the field. Coatings B and D suffered minor damage and prevented any deep erosion to occur. Coating A exhibited poor bond strength to the concrete substrate. Coating C did not exhibit sufficient resistance to cavitation erosion resulting in deep erosion to underlying concrete backfill.

MAINTENANCE OF COATINGS

It is apparent that under the current gate operating procedures, there is a maintenance requirement each year to repair the coatings. Fortunately, the maintenance should be minimal for coatings B and D because there was no deep erosion in the underlying concrete backfill. The repair time for these coatings should be only a few hours for one person.

CONCLUSIONS

This study has shown that protective coatings can be successfully applied to concrete hydraulic structures prone to cavitation erosion. The major conclusions were:

1. Performance of the coatings applied in the field showed that polymeric coatings can significantly reduce the erosion resulting from cavitation.

Figure 5: Coating B after application

Figure 6: Damage to Coating B after 1 year. Minor damage in one corner.

2. The use of polymeric coatings to repair damaged areas will require some maintenance. However, these repairs can be performed very quickly with relative ease.

3. It is very important that polymeric coatings have high bond strengths to the concrete substrates.

4. Laboratory tests showed that the thicknesses of polymeric coatings has to be controlled in order to be compatible to the concrete substrate.

5. Minor changes to the current gate operating procedures may eliminate or significantly reduce the need for maintenance of the cavitation resistant coatings.

ACKNOWLEDGMENT

The authors gratefully acknowledge BC Hydro and also the Canadian Electrical Association for funding this research program.

REFERENCES

Banks, P.A. Cass, D.E., Yung, K.Y.C., "The effect of Spillway Operation on Cavitation Damage to Hugh Keenleyside Dam Spillway". Hydraulic Engineering - Proceedings of the 1990 National Conference, San Diego, CA, USA, Jul 30-31, 1990

Cass, D., Banks, P.A., Yung, K.Y., "Hugh Keenleyside Dam Cavitation Damage on Spillway", B.C. Hydro Hydroelectric Engineering Division, Hydrotechnical Department, Internal Report No. H1922, March 1987.

Cheng, C.L., Webster, C.T., Wong, J.Y., "Reduction of Cavitation Erosion Damage on Hydraulic Structures Through The Use of Coatings", Contract No. 511 G530, Canadian Electrical Association, October 1987

Schrader, E., Tatro, S., "Cavitation and Erosion Damage to Concrete in Dams", Concrete International, March 1987.

NEW LOW COST SURFACE PREPARATION PROCESS FOR RELINING EXISTING PENSTOCKS

Richard D. Stutsman[1], M. ASCE

ABSTRACT

Pacific Gas and Electric Company (PG&E) has 72 hydroelectric power facilities located throughout its distribution system. Each of these facilities has at least one penstock, which conveys water from a dam, forebay, intake structure, tunnel, or canal to a powerhouse. To preserve the structural integrity of these steel penstocks, corrosion protective linings and coatings were initially provided.

However, due to the age of many of these penstocks, the lining materials have deteriorated and the penstocks are beginning to show severe corrosion damage.

When looking at the cost of relining penstocks, approximately 50% to 70% of the total cost is attributed to removal and disposal of the old lining materials and providing surface profile. Historically, the accepted process for removing old linings and rust scale, and providing surface profile has been by abrasive blasting to a Steel Structure Painting Council Cleaning Standards SSPC SP10, "Near-White Condition" and obtaining a 3-4 mil surface profile. This criteria is to assure that adequate adhesion strengths can be achieved with the new lining material.

PG&E has determined that by using a 10,000 psi waterblast system alone, the old lining materials and rust scale can be removed, and adequate lining adhesion strength can be obtained from the corroded steel surface. Although water blasting does not create a surface profile, it has been demonstrated that the depth and density of corrosion pitting can result in adequate surface roughness.

Waterblasting can reduce surface preparation costs by as much as 50% and significantly reduce the overall facility outage time.

1 Senior Civil Engineer, Pacific Gas and Electric Company, 1 California Street, F-1735, San Francisco, California 94111.

INTRODUCTION:

General: Pacific Gas and Electric Company (PG&E) has 72 hydroelectric power facilities located throughout its distribution system. The locations of these facilities extend from the Pit River, near Redding in Northern California, to the Kern River, near Bakersfield in Southern California. The average age of the facilities is over 50 years, with the oldest beginning commercial operation in 1890 and the most recent beginning operation in 1987. The hydraulic head on these facilities varies from 60 feet to 2558 feet. As such, they are a diverse mix of high and low head, old and modern structures.

Each of these facilities has at least one penstock, which conveys water from a dam, forebay, intake structure, tunnel, or canal, to a powerhouse. The majority of the penstocks are steel construction.

Background: To preserve the structural integrity of the steel penstocks, corrosion protective linings and coatings were initially provided. The majority of these original lining materials were coal tar enamels.

However, due to the age of many of these penstocks, the lining materials have deteriorated and the penstocks are beginning to show signs of severe corrosion damage.

When looking at the cost of relining penstocks, approximately 50% to 70% of the total cost is attributed to removal and disposal of the old lining materials and providing surface profile. Historically, the accepted process for removing old linings and rust scale, and surface profile has been by abrasive blasting to a Steel Structure Painting Council Cleaning Standards SSPC SP10, "Near-White Condition" and obtaining a 3-4 mil surface profile. This criteria is to assure that adequate adhesion strengths can be achieved the new lining material.

PG&E has determined that by using a 10,000 psi waterblast system alone, the old lining materials and rust scale can be removed, and adequate lining adhesion strengths can be obtained from the corroded steel surface. Although water blasting does not create a surface profile, it has been demonstrated that the depth and density of corrosion pitting can result in adequate surface roughness which will provide a substrate that provide that necessary adhesion strengths. Adhesion testing on numerous samples of corroded penstock sections, using nine 100% solids epoxies and urethanes, has resulted in adhesion strengths in excess of 1,000 psi. In fact, in some cases, tests using abrasive blasting of some corroded plate samples has resulted in lower adhesion strengths due to reduction of the overall surface roughness.

Because these penstocks are connected to operating facilities, access for any blasting process and removal of the byproduct can be difficult and costly to perform. For the

abrasive blasting operation, many tons of sand, coal tar residue and iron oxide remanents must be manually removed and disposed of as a hazardous waste. However, for the waterblasting operation, the blast water, residual coal tar and iron oxides can be pumped out and filtered; and the water can be recycled. Because no sand medium is used, only small amounts of the coal tar residue and iron oxide remanents need to be disposed of as hazardous waste.

The benefits of waterblasting can reduce the overall surface preparation costs by as much as 50% and significantly reduce the overall facility outage time.

Waterblast System: PG&E has developed and constructed a medium pressure, remote-operated, waterblasting machine. It can apply water at volumes of up to 300 gallons per minute and of pressures up to 10,000 psi. The water is discharged through multiple nozzles (up to 8) using pencil stream type nozzles which are located around the circumference of the rotating head. The water driven pump rotates the blast head at approximately 24 to 32 RPM. The blast head assembly is attached to a carrage beam outfited with four centering wheels at each end. The carriage beam length is adjusted for each specific project such that it is at least longer than the pipe diameter. The waterblast machine is then pulled through the length of the penstock by a diesel operated cable hoist. The blast water is supplied from a high-pressure, high volume pump truck to the rear of the waterblast machine through a 1-inch-diameter high pressure line. The waterblast machine is pulled through the penstock at approximately 1.5 linear feet per minute.

This blasting system and conventional coal tar epoxies have been sucessfully used for relining existing penstocks by PG&E for approximately 10 years. However, verification of how effective the process is and what its limitations are has not been demonstrated. Therefore, the test program described below was developed and implemented.

COATING TEST PROGRAM:

Coating Selection: Nine commercially available coating materials were selected from a broad cross section of manufacturers. All materials were from two generic chemical classifications; epoxies and polyurethanes. Epoxies have been known as coating "work horses" showing tenacious adhesion as corrosion resistant linings. However, they tend to be brittle in nature and do not cure at low temperatures which may be prevalent during field application. On the other hand, polyurethanes generally have good abrasive and elastomeric properties A flexable coating becomes very important when used on riveted penstock joints which will expand and contract with temperature changes. Polyurethanes also cure at low temperatures in less time than most epoxies offering added economic benefits.
The ultimate goal of this testing program was to develop a list of solventless epoxy and polyurethane materials that could be applied primerless (against bare metal) in a

single coat application, and could develop adhesion strengths in the 1,000 psi range. The following lists the materials tested and the abbreviations that are used later in this paper.

EPOXIES	**POLYURETHANES**
Carboline 163-2 (CARB)	Elastuff 160 (EL60)
Dampney I-782 (DAM)	Polybrid 705 (POL)
Enduraflex 1955 (EF)	
Epicon 329 (EPC)	
Irathane 155 (IRA)	
Madison Corropipe 17151 (MAD)	
Techthane 9055 (TEC)	

The materials under evaluation were applied according to the manufacturer's recommended instructions. When primers were recommended, the sample was prepared using the manufacturer's specified primer for one sample and without primer for an additional sample for comparison. A minimum of four samples were prepared for each material selected. They were applied over plate samples from two different penstocks prepared in two different ways. Two samples were abrasive blasted and two were waterblasted. All coating materials were applied to a 30 mil dry film thickness.

Steel Penstock Sample Selection: A total of 76 different steel samples, believed representative of actual corrosion conditions present in most of the penstocks, were selected from two hydroelectric facilities which were removed during retrofit work. Sample characteristics are as follows:

Caribou Penstock:

- 1921 vintage, riveted construction, hot-rolled carbon steel, no evidence of remaining mill scale, approximately 1" wall thickness.

- Substrate exhibits wide spread deposits of corrosion products as shown in SSPC Pictorial Surface Preparation Standards for Painting Steel Surfaces "Guide To Visual Standard No. 1" Rust Grade D. Substrate pitting to 0.250 inches in depth. Heavy rust scale present prior to surface preparation.

Angels Penstock:

- 1950 vintage, welded construction, hot-rolled carbon steel, visual evidence of minor deposits of previously applied coating of coal tar and intact mill scale was observed, approximately 1/4 " wall thickness.

- Substrate exhibits wide spread deposits of corrosion products as shown in SSPC Pictorial Surface Preparation Standards for Painting Steel Surfaces "Guide To Visual Standard No. 1" Rust Grade D except where presence of previously applied coating. Substrate pitting to 0.125 inches in depth. Heavy rust scale present prior to surface preparation.

PENSTOCK SAMPLE SURFACE PREPARATION:

Abrasive Blast Method: A total of 36 samples were prepared as "control samples" by traditional abrasive blast methods according to SSPC-SP10 "Near-White Metal". Air pressures used for blasting were measured as 95-105 psi at the nozzle and a #6 (3/8) venturi type nozzle was used. Abrasives used were 20-50 mesh Green Diamond brand mineral slag. Anchor blast profiles were measured ranging between 3.0 to 3.2 mils.

Medium Pressure Water Blast Method: An additional 36 sample sections were prepared using fresh water and a hand held water blaster operating at a gauge pressure of approximately 10,000 psi and 6 gallons per minute. The blast equipment was fitted with a stainless steel #1503 tip. Surfaces were water blasted to visually remove all surface corrosion products and provide a uniform visual metallic luster. Rust stain in the metal was permitted to remain. The anchor profiles resulting from corrosion (waterblast at 10,000 psi only cleans and does not etch the metal) ranged between 4.0 to 5.0 mils for the Caribou Penstock and 2.9 to 3.2 mils for the Angels Penstock samples. It should be noted that the pitting for the Caribou samples was a very dense pattern, where the pitting from Angels was only localized. All prepared surfaces were coated the same day.

An additional 8 sample sections were prepared in the same manner and allowed to weather 3 to 4 days before coating for evaluating the effects of adhesion under prolonged weathering conditions to simulate a delay in coating operations. Weathered samples did receive a fresh water rinse at approximately 5,000 psi before coating.

COATING APPLICATION:

Due to the many and varied materials selected for the evaluation, most of the methods of spray application were represented including traditional conventional spray equipment (mostly primer application), airless spray equipment (long pot life

coatings) and plural component spray equipment (multi-component solventless coatings). Plural component equipment also uses specialized equipment capable of heating (to reduce material viscosity's in lieu of solvents and control component reaction rates), proportioning (pumps specifically meter amounts of each component) through individual, insulated and heat traced, high pressure paint lines and mixed in line (combines and thoroughly blends) for spray application.

PENSTOCK COATING ADHESION EVALUATION - TEST PROCEDURE:

Two separate adhesion tests were performed: one soon after the initial cure of the coating material, and a second one after 90 days of immersion in water. The immersion test is designed to test the permeability of the coating and its effect on adhesion.

Coating adhesion tests were performed on each type of coated penstock sample in accordance with ASTM D-4541 "Pull Off Strength of Coatings Using Portable Adhesion Testers." The Elcomoter brand tester was used since many companies in the coating industry are familiar with this test and considerable adhesion test data is currently available for this test method.

Three pre-blasted adhesion dollies per sample type were installed to perform the tests. The dolly anchor profile was between 2.1 and 2.3 mils. Araldite Epoxy Structural Adhesive was used for bonding dollies to the coating surface and 72 hours were allowed for cure before performing the test. The substrate temperatures ranged between 63 and 68 degrees F during cure time.

Adhesion dollies were bonded to the high points of the plate to avoid the uneven pitted surfaces. This should result in conservative (lower) values of adhesion.

The procedure started with a 0-1000 psi tester and continued to the 0-2,000 psi tester until the dolly released. A pull stress value and the method of failure were recorded for each failed dolly. Methods of failure are described in the ASTM test method as Glue Failure, Cohesive Failure and Adhesive Failure and the percentages attributed to the failure type, should more than one failure type occur. Glue failure is a release of the dolly from the coating surface due to the adhesive bond. Cohesive failure occurs when the coating pulls apart within itself or from the primer. Adhesive failure occurs when the coating pull away from the penstock substrate.

COATING ADHESION EVALUATION - PRE-IMMERSION TEST:

The following five results were identified from the pre-immersion tests:

1) Coating adhesion values for a broad cross section of coatings were consistent between tests of the same material and within generic coating types when applied

over visually cleaned high pressure waterblasted steel surfaces with measurable corrosion surface profile.

2) Coating adhesion values for visually cleaned medium pressure waterblasted steel surfaces compared quite favorably to the abrasive blasted surfaces (SSPC-SP10) used as control surfaces. Some coating materials applied over the waterblasted surface actually exceeded the values obtained for the abrasive blasted surface for all three tests.

3) Surface profile is important in obtaining high coating adhesion values whether derived as a product of corrosion or abrasive blasting. A 4.0 to 5.0 mil measured surface profile from the corrosion pitting produced consistently higher adhesion values than the measured 3.0 to 3.5 mil abrasive blasted surface profile.

4) The method of failure during testing is important to note. Coating "Pull-Off Strength" values are a more correct description than coating adhesion values because the values themselves do not indicate whether the glue used to bond the dolly to the coating surface failed, the coating split within itself or the coating let go of the steel substrate.

5) It appears from testing that some specific coating and generic coating types have been identified as having a greater affinity to steel adhesively than others. In general, the chemical family of epoxies is recognized for this characteristic and was quite consistent in these tests either as a one coat system or where epoxies were used under epoxies or other chemical families (i.e., polyurethane's). Traditional two coat systems (primer/finish) did not really exhibit any advantages when compared to the new one coat solventless epoxies or polyurethane's applied hot with special plural component equipment.

COATING ADHESIVE EVALUATION - POST-IMMERSION TEST:

The following two results were identified from the post-immersion tests:

1) Coating adhesion values for both coating types remained quite consistent between pre-immersion and post-immersion tests of the same material and within generic coating types without regard to method of surface preparation. Specifically coatings that offered high "pull-off strength" values under pre-immersion conditions also offered relatively high values in post-immersion conditions.

2) Overall, post-immersion adhesion values continue to identify that the plural component epoxy coatings, as a generic coating type, having a greater affinity to steel without need for primers than their polyurethane's counterparts. The plural component epoxy materials applied using heat for viscosity control also appear to yield higher "pull off strength" values than other 100 percent solid epoxies (i.e., Dampney Coal Tar Epoxy) applied without heat.

CONCLUSIONS:

- The results indicate that for abrasive blasted surfaces, sufficient corrosion pitting is necessary to obtain acceptable adhesion strengths (i.e., Caribou corrosion versus Angels corrosion). See Figures 3-1 and 3-2.

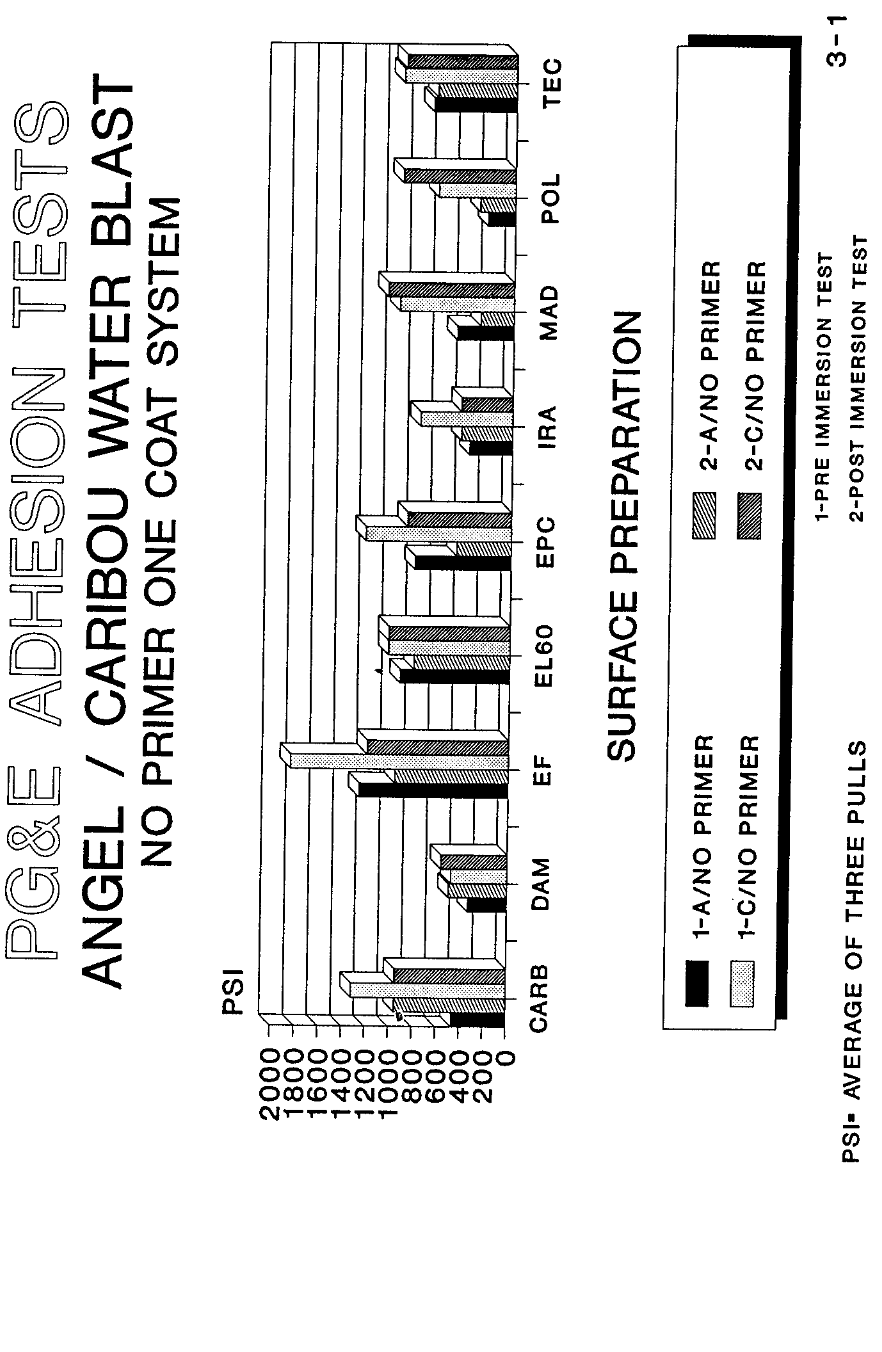
- Both the pre-immersion and post-immersion test results demonstrate that adhesion values for waterblast are consistant with samples prepared by sand blasting. In some cases, the waterblasted adhesion values exceeded the abrasive blasted values. See Figures 3-1 and 3-2.

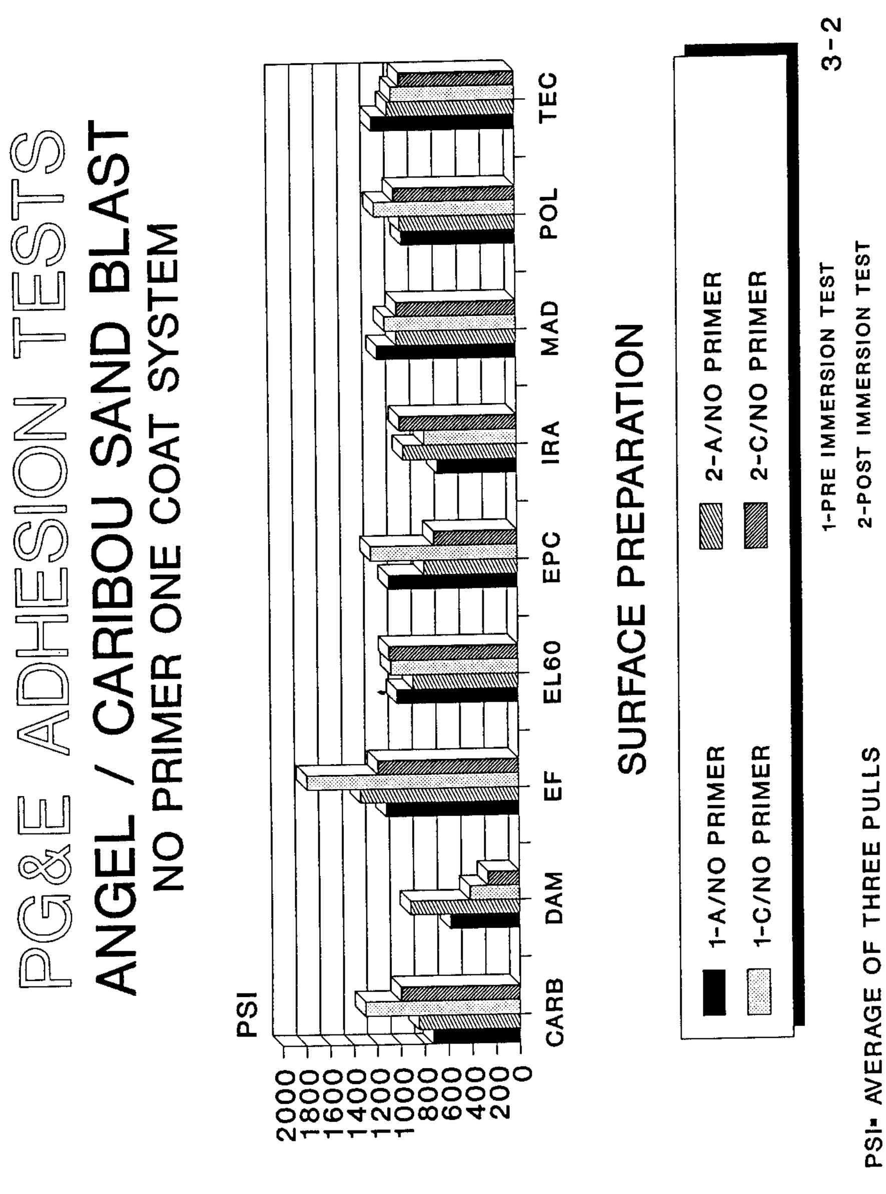
- Both the pre-immersion and post-immersion test results demonstrate that elimination of the primer for products that specify a primer, generally do not impact their adhesion strengths. In fact 4 out of the 5 materials specifing a primer showed higher adhesions strengths without the primer. The fifth had 20% lower adhesion strengths, however its strength was 950 psi.

- Both the pre-immersion and post-immersion test results demonstrate that all nine materials can be applied in a single coat application to 30 mils and not significantly impact the adhesion strengths of the materials.

- Based on the results of the waterblasted tests on the corroded Caribou Penstock, the following expoxies and polyurethanes have demonstrated acceptable adhesion strengths:

EPOXIES	POLYURETHANES
Carboline 163-2 (CARB)	Elastuff 160 (EL60)
Enduraflex 1955 (EF)	Polybrid 705 (POL)
Epicon 329 (EPC)	
Madison Corropipe 17151 (MAD)	
Techthane 9055 (TEC)	


PG&E ADHESION TESTS
ANGEL / CARIBOU WATER BLAST
NO PRIMER ONE COAT SYSTEM
PSI
2000
1800
1600
1400
1200
1000
800
600
400
200
0
CARB
DAM
EF
EL60
EPC
IRA
MAD
POL
TEC
SURFACE PREPARATION
1-A/NO PRIMER
2-A/NO PRIMER
1-C/NO PRIMER
2-C/NO PRIMER
1-PRE IMMERSION TEST
2-POST IMMERSION TEST
3-1
PSI= AVERAGE OF THREE PULLS


PG&E ADHESION TESTS
ANGEL / CARIBOU SAND BLAST
NO PRIMER ONE COAT SYSTEM
PSI
2000
1800
1600
1400
1200
1000
800
600
400
200
0
CARB
DAM
EF
EL60
EPC
IRA
MAD
POL
TEC
SURFACE PREPARATION
1-A/NO PRIMER
1-C/NO PRIMER
2-A/NO PRIMER
2-C/NO PRIMER
1-PRE IMMERSION TEST
2-POST IMMERSION TEST
PSI= AVERAGE OF THREE PULLS
3-2

Automation of The Osage Hydroelectric Plant

Steve L. Duxbury[1]
Robert W. Ferguson[2]

Abstract

In recent years, a large number of automation upgrades in hydroelectric power plants were made because of the owners' desire for plants to operate more efficiently by requiring fewer personnel for operation and maintenance of the equipment. Many older power plants will require replacement because of their age, high maintenance requirements, and lack of spare parts availability. Many older plants lack instrumentation and monitoring equipment, and the operator must manually monitor the unit to ensure that it is operating safely. Today, with the advance of electronic equipment, and specifically, the development of microprocessor based control systems, plants can be operated from remote locations, requiring less manpower for operation and maintenance. Since the plant can be operated at a higher efficiency by fewer personnel with this new equipment, the cost of the automation is returned in a reasonable amount of time. With increased monitoring and operating parameter setpoints, the plant can operate more safely. This paper describes a typical automation upgrade recently performed at the Osage Hydroelectric Plant.

---

[1]Engineering Manager, Black & Veatch, 11401 Lamar, Overland Park, Kansas 66211.

[2]Project Manager, Union Electric Co., 1901 Chouteau Ave., St. Louis, Missouri 63166.

Introduction

The Osage Hydroelectric Plant is owned and operated by the Union Electric Company. The plant is part of Bagnell Dam, which is located on the Osage River at Lakeside, Missouri. The lake impounds approximately 2 million acre feet of water. The plant was commissioned in 1931 and included six main units and two house units. In 1953, the plant was expanded to include two additional main units. The eight main turbine generator units are vertical shaft Francis type turbines rated 33,500 horsepower with the generator output rated at 21.5 MW each. The generators operate at 13,800 volts. The two house units are also vertical shaft Francis type turbines rated 3,025 horsepower. The total plant gross output capacity is rated at 212 MW, which is supplied to the grid at 132,000 volts. Each unit is equipped with an Allis Chalmers governor for speed control and loading of the units. Each pair of main unit governors shares a common governor oil pump, pressure tank, and sump. Each house unit governor has its own pump and reservoir tank. Excitation current for each main unit is supplied by motor driven exciters. Each house unit is equipped with a shaft driven exciter.

Prior to the automation upgrade, the units were manually started and stopped from control stations located near the turbine generators. The units were then synchronized by the operator in the central control room. Once the units were synchronized, they could be placed on area load dispatching so that the load could be regulated by remote raise and lower pulses. These pulses were received at the plant through a remote terminal unit which was interfaced with a Honeywell programmable logic controller (PLC). The Honeywell PLC, which replaced the original Leeds & Northrop equipment, was installed in 1989. The PLC compared the output of each unit and sent raise pulses to the unit with the lowest load output and lower pulses to the unit with the highest load output. This allowed the plant to balance the load among all of the on-line units. Voltage and var control were manually operated. Very few additional upgrades had been performed on the rest of the plant equipment since the plant was originally commissioned.

Automation Decision

Union Electric began exploring the idea of automation for the Osage plant in the mid-1980s because of rising maintenance costs and the desire to improve plant efficiency. The Osage plant is primarily operated as a peaking plant, but is also very important in Union Electric's grid as a back-up source of generation and voltage control. Because of the length of time it took to bring a unit on-line, it was necessary to keep the units running in a synchronous condensing mode so that the plant could

respond quickly to load demands on the grid. These concerns contributed to the decision to automate.

The next step was to determine which equipment to automate. Any automation decisions must be cost effective and provide reliable operation and improved efficiency. A decision was made to replace the manual controls in the central control room with a new digital control system which allows automatic startup, synchronizing, loading, and shutdown. It was also decided that the existing manual governor controls would be replaced with a PLC-based digital governor system due to their added benefit versus the relatively small increase in cost; however, the existing governor oil pumps, pressure tanks, and servomotors would be retained. The house unit governors would be replaced with digital governors but would also receive a complete new high pressure hydraulic system. To improve the reliability and the response time of the excitation system, the existing motor driven exciters would be replaced with new static excitation equipment, including automatic voltage regulators. The house unit shaft driven exciters would be retained, but would receive new automatic voltage regulators. Automatic synchronizing equipment would be added for both the main and house units. The main and house units had an existing manual gate lock system that would need to be automated.

A major addition to the automation project would be a new supervisory control and data acquisition (SCADA) system. This system provides overall control of the units from the central control room and handles the interface for the remote dispatching requirements.

Black & Veatch was retained to provide the services for engineering design and implementation. Black & Veatch will coordinate the design with the equipment manufacturers, provide specifications for the design and coordination of the SCADA system, design the construction drawings and specifications, and provide construction support services when required. This includes providing detailed designs to automate equipment such as the main and house unit gate lock system, the air compressor system, unwatering pumps, and a fire detection system for remote monitoring.

Automation Equipment Descriptions

Governors. Following an evaluation of the governor, Union Electric decided to include the local automatic operation of the units as part of the digital governor package, using an integrated governor/controller that would control the governor as well as provide unit automatic start/stop control. The contract for this equipment was awarded to Voith Hydro,

Inc. Voith provided a new hydraulic actuator assembly with a distributing valve as a replacement for the existing governor on the main units. Entirely new hydraulic systems, including governor pumps, tanks, and proportioning valves, were supplied for the house units.

Voith used their Voith Control Center (VCC), a PLC-based digital system, which provides automatic turbine control as well as local automatic start-stop control of the units. The VCC uses an Allen-Bradley 5/25 PLC to perform the control functions. The units can be entirely controlled locally from the VCC panel near each individual unit, allowing the units to be put on-line with a simple start command. The controller interfaces with the remote SCADA system (located in the central control room) through a data highway system for remote control of the units. Each VCC panel has pushbuttons, selector switches, indicating lights, and analog meters for local control and monitoring. The panel contains a video display monitor to display screens for monitoring and alarms.

Excitation System. The contract for replacement of the existing excitation system for the main units was awarded to Asea Brown Boveri (ABB). ABB provided a potential source, static type excitation system which included a power potential transformer and an automatic voltage regulator. ABB also provided automatic voltage regulators for each of the house units.

Automatic Synchronizers. Voith provided synchronizing panels which are located next to the VCC panels by each unit. Individual synchronizing panels were purchased which provide a complete stand-alone control system for each main unit. The two house units share a common synchronizing panel.

Gate Locks. The existing manual gate locks were automated to incorporate them into the automatic start/stop sequence of unit control. The existing gate lock system for the main units was a hinged bar mechanism which would manually lower onto the gate shaft of one of the servomotors. The new design required that the locking mechanism be removed from the hinge and mounted onto a vertical sliding mechanism, which would be put into position using a motor operator.

Miscellaneous Equipment Automation

In addition to automation of the primary unit equipment, upgrade and modification of some of the auxiliary equipment was required to provide the plant with full remote operation and monitoring capability. The existing unwatering pumps were manually started and stopped. As a part of the automation, level switches and control equipment were added to

alarm critical water levels in the unwatering sump and to automatically start the pumps. Manual controls to stop the pumps were retained. Each pump was equipped with an oil lubrication system requiring manual operator action to provide prestart lubrication and establish adequate oil flow for pump operation. These lubrication systems were replaced with systems which automatically provide the prestart and operational lubrication oil flows and alarm low oil level in the oil reservoirs.

The compressed air supply to the governor accumulator tanks was also automated. Before automation, an operator was dispatched once per shift to check the air pressure in each tank. If the pressure was low, the operator would open a manual valve and allow additional air into the tank to bring the pressure to the required value. The system was automated using the new SCADA system and the pressure and oil level in each tank is now monitored. A program was developed to compare the actual pressure in the tank with the desired pressure at the actual oil level. If the pressure is low, a solenoid valve in the air supply line to the tank is opened to allow air into the tank until the pressure reaches the correct value.

The existing fire detection system monitored only the turbine generators. The new detection system includes other areas in the plant to provide more comprehensive monitoring and notification of possible fire conditions.

SCADA

One of the major design considerations for the automation project was the SCADA system. The system needed to accomplish the following criteria:

- Provide total plant control from the control room.

- Interface with the individual unit VCC panels using a data highway system.

- Interface with the remote dispatch terminal unit.

- Provide balance-of-plant data acquisition using remote Input/Output (I/O) panels.

- Include a program for var/voltage control of the plant and units.

- Include either a new program for load control or interface with the existing Honeywell load controller.

- Provide alarms to replace the existing annunciator.

- Provide data logging and archiving.

An evaluation determined that the existing Honeywell load controller should not be reused. It was decided that this function would be incorporated in the station PLC program as part of the SCADA design. Bids were sent to various custom distributed control system and micro-computer based system vendors. The microcomputer based system was chosen because of the economic advantages. The contract was awarded to Durkin Equipment Company, Inc.

The Durkin system is based around an IBM 7561 computer system. A diagram of the overall control system is shown on Figure 1. The PLCs are linked to the control system using redundant A-B 1785-KE communication interface modules. Communication between these modules and the Voith VCCs is accomplished using five twisted pair data highways:

- Highway 1 - House Unit 1 and Main Unit 1.

- Highway 2 - House Unit 2 and Main Unit 2.

- Highway 3 - Main Unit 3 and Main Unit 4.

- Highway 4 - Main Unit 5 and Main Unit 6.

- Highway 5 - Main Unit 7 and Main Unit 8.

The two communication modules are also linked to two IBM 7561 computer work stations, located in the control room, through RS-232 interface cables. One computer is the main running unit and the other is in hot standby. There are three additional IBM 7561 computer work stations which are networked, with the two main stations using a Quantum LAN active hub. Each work station includes the following equipment:

- IBM 7561 Computer.

- IBM 7554 Graphic Display Monitor.

- IBM Keyboard.

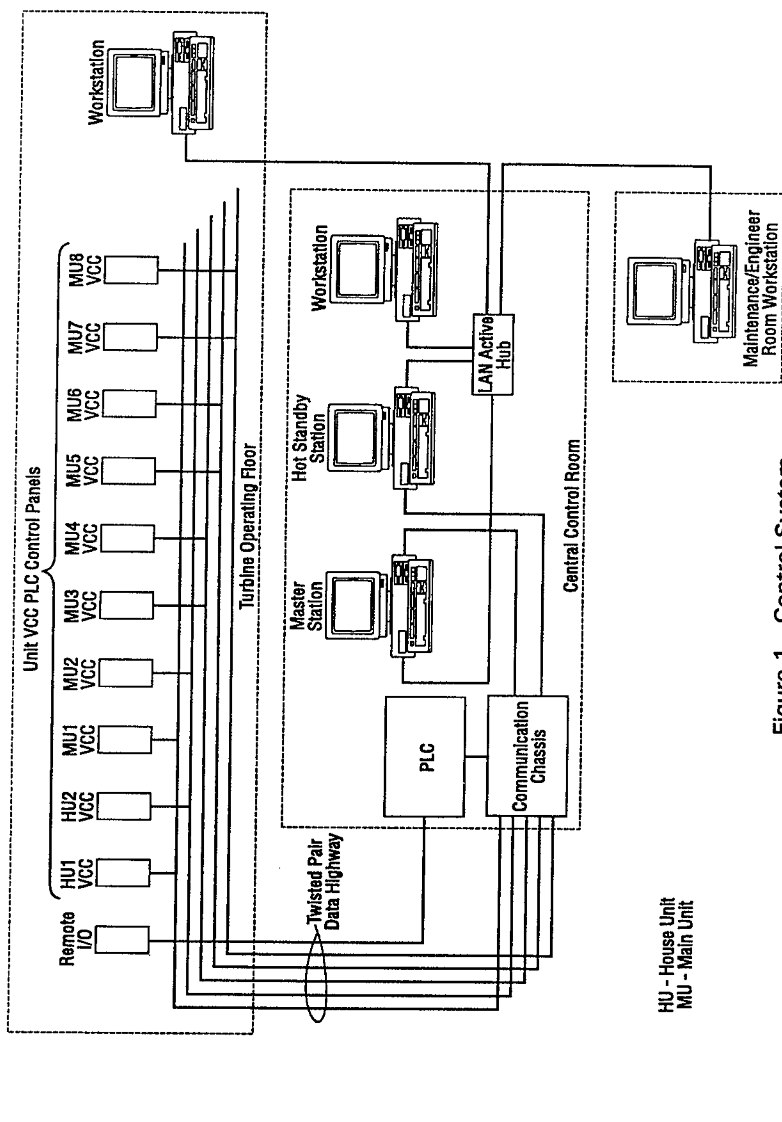


Figure 1. Control System

- A-B Advisor Intelligent Control Panel.

- Printers (three work stations).

- Optical Disk Drive (one work station).

Three of the work stations are located in the control room and provide complete control and data acquisition capability for the entire plant. One work station is on the turbine floor and is identical to the control room work stations; however, control of the units is only allowed with special password permission. The other work station, in the Maintenance/Engineering office, does not have provisions for control of the units.

The station PLC is an A-B PLC 5/25. This PLC is identical to the PLCs located in the VCC panels. This PLC contains a load program that replaces the program in the existing Honeywell load controller which receives raise/lower commands from the dispatch center and sends these commands to the units which are on-line. A raise pulse setpoint is sent to the lowest loaded unit and a lower pulse setpoint is sent to the highest loaded unit. When a unit is taken off-line, the program must again balance the units. The existing load program transmits the entire raise/lower load request to a single unit unless that unit has reached its limit. If so, it then transmits the remaining load value to the next appropriate unit. The new program was upgraded to send the pulses in 1.4 MW increments, which improved the capability to balance between the units. The SCADA PLC also has a voltage control program that interfaces with the excitation system to send raise/lower setpoints to the automatic voltage regulators based on voltage setpoints which are input by the plant operator for each of three buses. There are four main step-up transformers, each of which connects a pair of units to the buses. The program must send the raise/lower pulses to the units which are on the bus which is being adjusted, and must also balance the var contribution between the pair of units.

The new control system has the ability to meet Union Electric's requirement to "quick" start the unit(s). If Union Electric has an emergency power demand requirement, units can be selected on the SCADA system screen and quickly synchronized and loaded to their 85 percent gate rating. For normal operation, unit(s) can be selected, synchronized, and loaded to a minimum 10 MW load level.

Construction

Construction for the automation project began in November 1991, with the House Unit 2 the first unit selected for automation. Because of the Fall 1992 delivery schedule of the SCADA equipment, it was necessary to allow the units to operate locally from the VCC panel as they were automated and put back on-line. Union Electric performed the construction for the house units. The contract for construction of the main units was awarded to Sachs Electric Company. Although unit construction was based on pairs of units due to common governor tanks, only one unit was taken off-line at a time. The major construction work was halted during the summer months to make all the units available. A smaller crew installed raceway and cable until the construction could resume in the fall. At present, all units have been automated and are operating from the Voith VCC panels. The plant control will be transferred to the new SCADA system shortly.

Automation: A New Source of Power for the Future

With the advent of low cost, reliable PLC-based control systems, a great number of existing hydroelectric plants are being automated to replace older, unreliable, less efficient manual control systems. There are a tremendous number of options as to the level and amount of automation that can be accomplished, depending on the owner's requirements. The cost to automate older hydroelectric plants can show a payback in a relatively short period of time. By increasing the amount of monitoring and alarming of vital functions, many power plants can operate unmanned or with a reduced staff. Combined with upgrade alternatives, older hydroelectric plants can continue to provide a reliable source of power for the future.

SWITCHGEAR FAULT, DAMAGE CONTROL, RECOVERY AND REDESIGN AT THE JOHN DAY POWERHOUSE

Bob Luck and Chuck Rinck [1]

ABSTRACT

On May 29th 1990 a fault occurred in one of the two station service switchgear lineups at the John Day powerhouse. The problems encountered due to the electrical failure of the station service switchgear are presented and the recovery procedures taken to restore operations from black start conditions are described. Results of the forensic and safety investigations conducted immediately after powerplant operations were restored are summarized. The interim system configuration, new equipment upgrade, and redesign of the station service system to improve protection and emergency response are also developed.

PROJECT DESCRIPTION

John Day Lock and Dam is a multi-purpose project located on the Columbia River, approximately 112 miles east of Portland, Oregon. The project is a run-of-the-river hydroelectric power plant. With its installed capacity of 2,160 megawatts the sixteen unit powerplant is the third largest in the United States. Besides producing power, the other project functions involve flood control, irrigation, slackwater navigation, fish passage, and water-related recreational facilities.

----

[1] Electrical engineers
U.S. Army Corps of Engineers
P.O. Box 2870, Portland, OR 97208-2870

The power generated is delivered at the hub of an extensive 500 kV transmission network. This site is at the northern end of ac/dc interties to southern and central California and is interconnected to the systems of British Columbia and Western Area Power Administration. Because of the dam's strategic location, large capacity, and system operational characteristics, the Bonneville Power Administration (BPA) has designated John Day as the first powerplant to be brought back on line to restore the northwest power pool's system.

Figure 1 shows the main one line diagram for the powerplant with it's 16 generator units connected in groups of four to deliver power through four 500 kV lines. Figure 2 is the one line diagram of the 4,160 volt station service electrical power distribution system for the project complex. Electrical power for station service is developed by a connection to the first four generator units through a current limiting reactor and a primary circuit breaker to a 13.8 kV/4.16 kV, 5.4 MVA transformer. A second source is obtained in a similar manner with connection of a redundant equipment lineup to the second set of four units. A modernization program under construction and nearing completion at the time of switchgear failure included a 4,160 volt, 750 kW emergency diesel generator as one of the powerplant improvements. Extensive rework to the controls, relaying, and metering within the station service switchgear was also underway as a part of this rehabilitation work when the switchgear failed.

The redundant 4.16 kV station service switchgear distributes power to auxiliary equipment for the main unit generators and for control power and general station use. This equipment had over 20 years service and contained obsolete air-magnetic type circuit breakers. The two lineups, designated SP1 and SP2, each had 12 air-magnetic type circuit breakers with associated current and potential transformers, overcurrent relays, control switches, and indicating lamps.

SWITCHGEAR FAULT AND RECOVERY

At 5:17 PM on May 29th of 1990 all station service power was lost. The plant outage was the result of an arcing fault within the SP2 switchgear. Only four station operators were on duty at that time since the normal day shift ended at 4:00 PM. One of the station operators discovered the arcing while walking toward the station service equipment on a routine check. A second operator nearby rushed over to help the first operator control the fire with CO2 type extinguishers. A third operator performing external powerhouse checks opened the spillway gates to maintain normal river flow. The main plant operator in the smoke filled control room on one of the upper levels stayed until the project was shut down in an orderly fashion before evacuating the

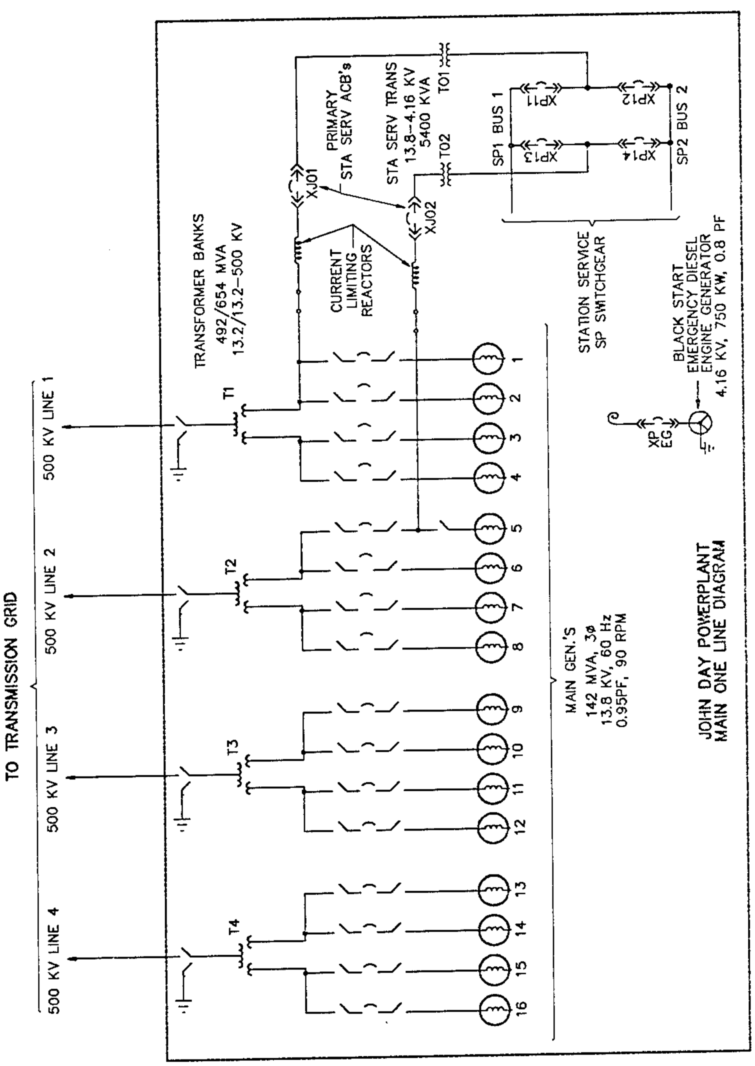
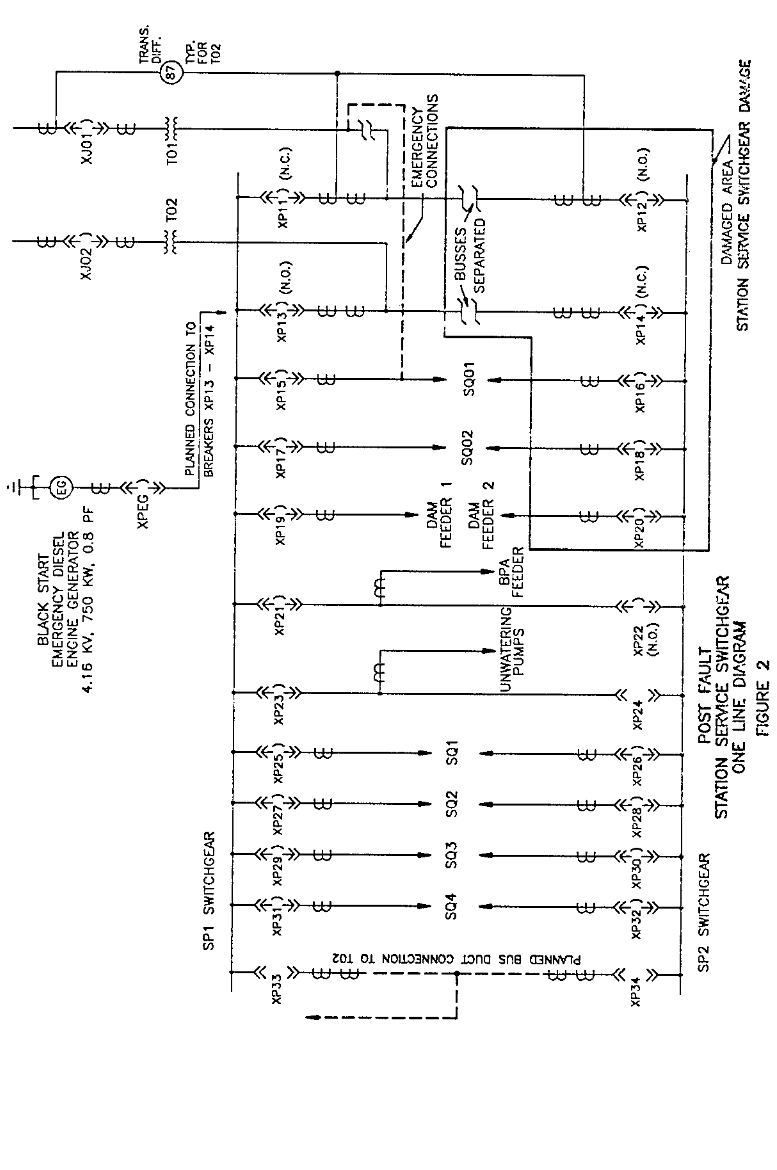

TO TRANSMISSION GRID
500 KV LINE 1
500 KV LINE 2
500 KV LINE 3
500 KV LINE 4
T1
T2
T3
T4
TRANSFORMER BANKS
492/654 MVA
13.2/13.2-500 KV
CURRENT
LIMITING
REACTORS
XJ01
XJ02
PRIMARY
STA SERV ACB's
STA SERV TRANS
13.8-4.16 KV
5400 KVA
T01
T02
SP1 BUS 1
SP2 BUS 2
XP11
XP12
XP13
XP14
STATION SERVICE
SP SWITCHGEAR
BLACK START
EMERGENCY DIESEL
ENGINE GENERATOR
4.16 KV, 750 KW, 0.8 PF
XP
EG
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
MAIN GEN.'S
142 MVA, 3ø
13.8 KV, 60 Hz
0.95PF, 90 RPM
JOHN DAY POWERPLANT
MAIN ONE LINE DIAGRAM

FIGURE 1



POST FAULT
STATION SERVICE SWITCHGEAR
ONE LINE DIAGRAM
FIGURE 2

powerhouse. Off duty staff members were then recalled to the site for additional support.

Significant damage was sustained to four of the twelve circuit breaker cabinets in the SP2 switchgear with the two station supply breakers and the bus duct connections at one end of the lineup having the most severe fault damage (see Figure 3). Other equipment nearby also suffered extensive heat damage.

A complete power outage, flooding of the turbine pits, a halting of the river navigation system, and stoppage of the fish attraction water pumps were the major effects of the equipment failures. Recovery procedures included temporary bus duct disconnections to the damaged switchgear and the use of mobile diesel engine generators. Emergency cable connections (see Figures 2 & 3) were made to regain a limited power source for working around the switchgear area. After cleaning off the smoke contamination and the successful insulation check of the SP1 switchgear, reconnections of the bus duct restored normal station service power. This then allowed pumping out the flooded lower galleries and eventual cleanup of the flooded generator turbine pits to restore normal plant operation.

Two station operators suffered from smoke inhalation even though self-contained air breathing apparatus were used. Remote hand held radio communications was hampered by the apparatus air masks. Many portable fire extinguishers were expended in the attempt to put out the fire which was sustained by the arcing for over 15 minutes.

An electrician was shocked resulting in minor injury to his right shoulder while cleaning up the smoke residue inside the undamaged SP1 switchgear to prepare for its reenergization. This accident occurred because the temporary backfeed connection was overlooked during the urgency of recovery. The electrician returned to work after two days of treatment and observations.

INTERIM OPERATION

Prior to beginning any extensive repair the powerhouse was capable of generating at full capacity with just the original SP1 station switchgear in service (see Figure 4). The fault damaged SP2 switchgear was disconnected from normal sources with its eight surviving breakers left in place ready for contingency backup service. The new 750 kW, 4160 volt black start diesel generator physically installed in the powerplant but not connected at the time of the fault was then connected to SP1 for standby service. Two temporary mobile diesel generators were located outside the powerhouse for use as emergency sources to provide power for the critical unwatering pumps and the generator turbine pit pumps.

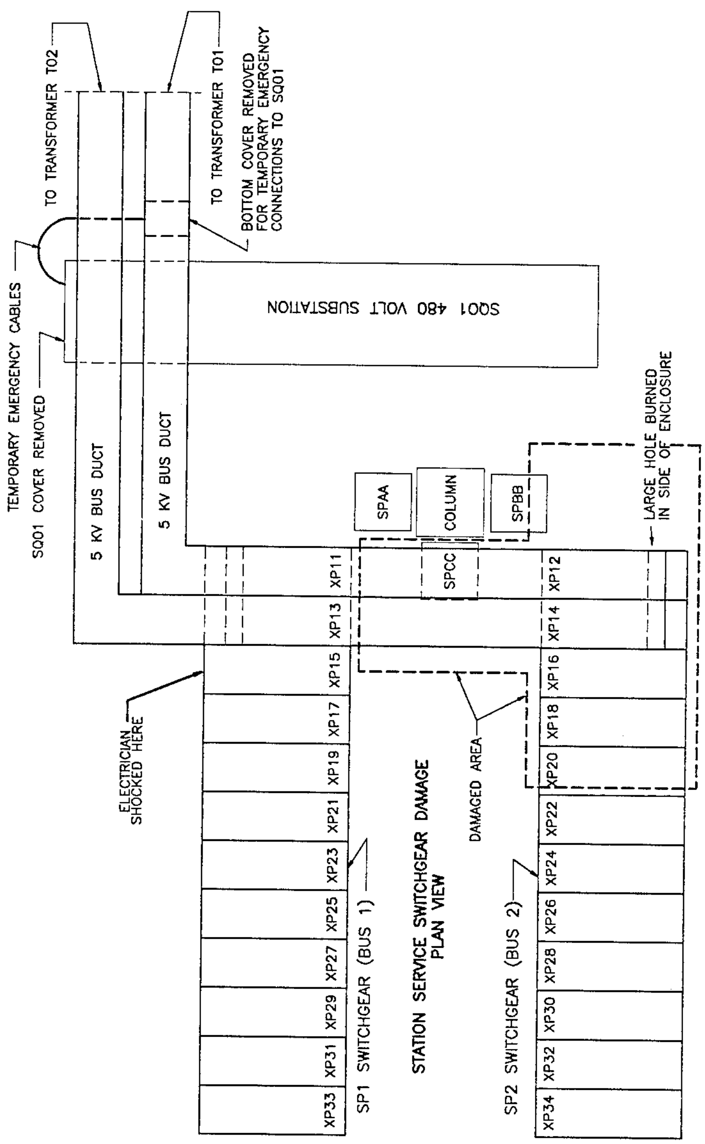
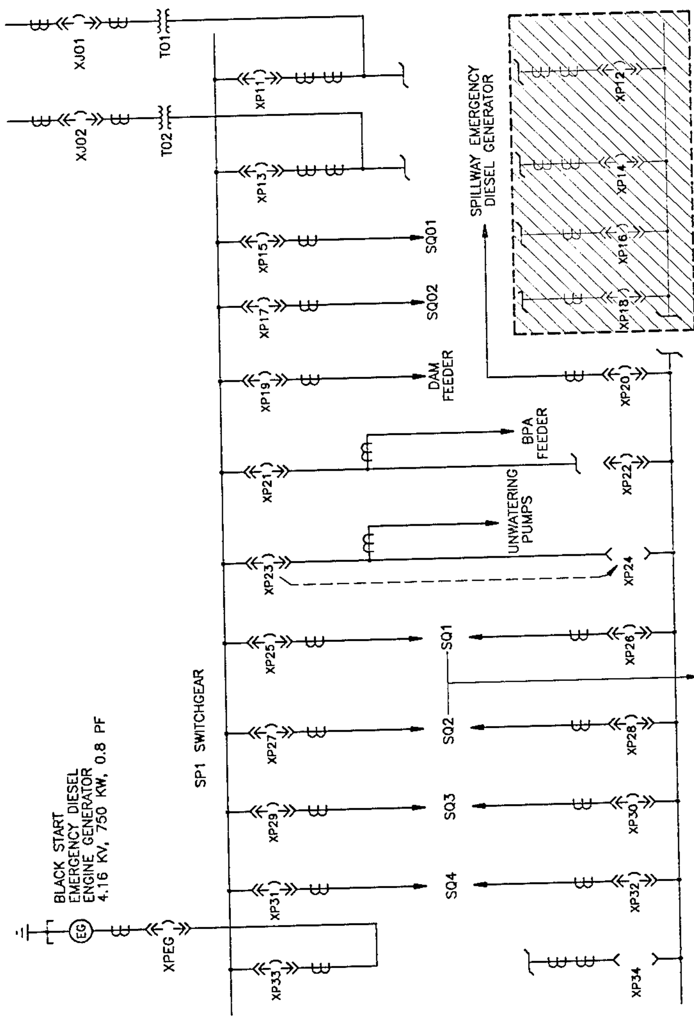

TEMPORARY EMERGENCY CABLES
SQ01 COVER REMOVED
TO TRANSFORMER T02
TO TRANSFORMER T01
BOTTOM COVER REMOVED FOR TEMPORARY EMERGENCY CONNECTIONS TO SQ01
SQ01 480 VOLT SUBSTATION
5 KV BUS DUCT
5 KV BUS DUCT
SPAA
COLUMN
SPBB
SPCC
LARGE HOLE BURNED IN SIDE OF ENCLOSURE
ELECTRICIAN SHOCKED HERE
XP11
XP12
XP13
XP14
XP15
XP16
XP17
XP18
XP19
XP20
XP21
XP22
XP23
XP24
XP25
XP26
XP27
XP28
XP29
XP30
XP31
XP32
XP33
XP34
DAMAGED AREA
SP1 SWITCHGEAR (BUS 1)
SP2 SWITCHGEAR (BUS 2)
STATION SERVICE SWITCHGEAR DAMAGE
PLAN VIEW

FIGURE 3



JOHN DAY LOCK & DAM
INTERIM STATION SERVICE
ONE LINE DIAGRAM

FIGURE 4

REPAIR PROGRAM

After restoration of powerplant operation, several repair alternatives were considered. These ranged from a minimum effort by simply replacing damaged circuit cubicles and bus duct sections to a more complete program with either a total retrofit or replacement of all the switchgear. Since the existing switchgear had more than 20 years service and contained obsolete air-magnetic type circuit breakers, it was decided to proceed with a complete replacement with state-of-the-art, two-high, metal-clad, sulfur hexafluoride gas (SF6) type switchgear.

SYSTEM AND EQUIPMENT CONFIGURATION DESIGN CHANGES

The changes to the switchgear and station service system configurations to upgrade with modern technology equipment and to provide increased protection with quicker recovery response include (see Figure 5):

- SF6 type circuit breaker switchgear

- Physical separation of the two normal station service source and emergency diesel generator positions in the switchgear lineup

- Bus duct fire and vapor barriers and disconnect links

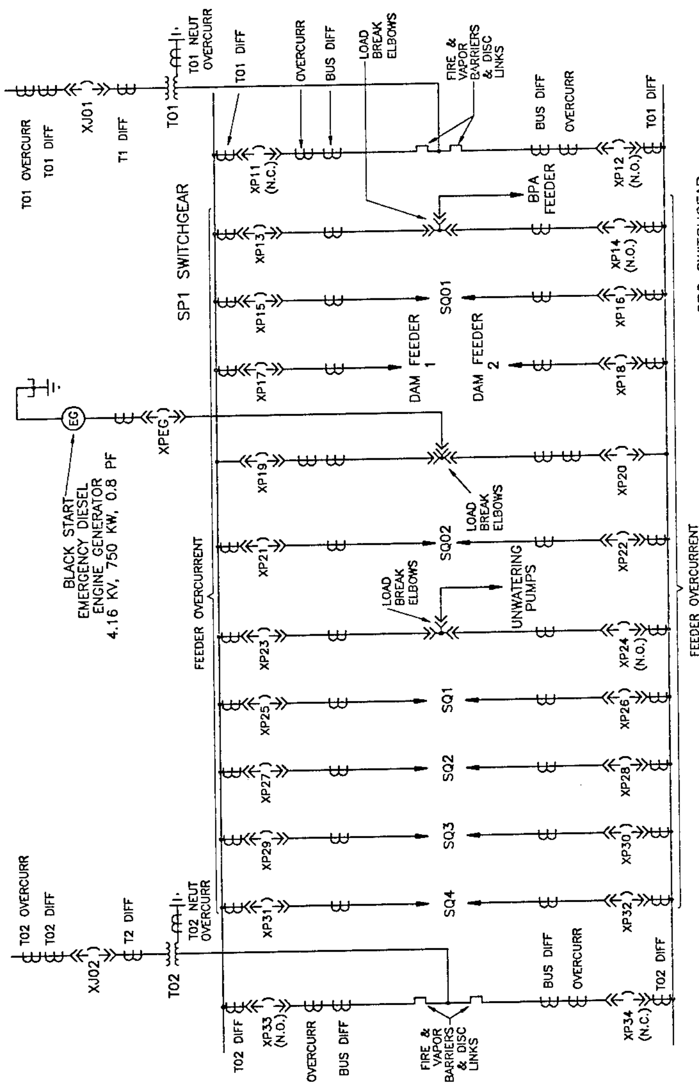
- Sectionalizing load break type elbows for the SP1 and SP2 common cable connections

- Switchgear main bus differential relays

- Backup primary breaker overcurrent relaying

REPLACEMENT SEQUENCE

Initial demolition work included the disconnection, removal, and disposal of the fault damaged SP2 circuit breaker cabinets and bus duct. The eight salvageable SP2 breaker cabinets were relocated to an adjoining location for possible reconnection during the reconstruction process. The new SP2 switchgear was installed in the space cleared and placed in service before the demolition workon the existing SP1 switchgear for installation of the new SP1 gear. The installation of the new switchgear which was performed by the project staff required extreme caution since the sequence of the various construction work elements had to be planned and controlled carefully in order to maintain continuous plant generation without any scheduled outages or accidental disruption.



REDESIGNED STATION SERVICE
ONE LINE DIAGRAM
FIGURE 5

FORENSIC AND SAFETY INVESTIGATIONS

A forensic investigation performed by a private engineering firm did not pinpoint an exact cause of the fault because of the extensive damage to the switchgear. Severe phase-to-phase and phase-to-ground arcing faults occurred in the bottom rear compartments of the station service source breakers in SP12 and SP14 which contain the main horizontal bus and the removable breaker connections.

An internal safety investigation was also immediately conducted after the outage. The temporary electrician that received an electric shock was working with clearance procedures established by one of the permanent staff electricians under the crisis conditions. The temporary electrician did not test the SP1 switchgear he was assigned to clean. During the recovery SP1 had been energized by the temporary emergency cable connections to activate the adjoining 4,160 volt-480 volt SQ01 substation for work lighting. The electrical backfeed from SQ01 to SP1 was overlooked in the urgency to halt the powerhouse flooding in process.

Procedures now require that each electrician whether permanent or temporary is held responsible to test for safe conditions before proceeding with any service activity. Air breathing apparatus has been upgraded for better fit and increased capacity. The fire protection and smoke control procedures were revised to incorporate the lessons learned based on the experience of this event.

VARIABLE SPEED IN HYDRO POWER GENERATION UTILIZING STATIC FREQUENCY CONVERTERS

Dr. L. Terens[1] and R. Schäfer[2]

ABSTRACT

Varspeed has already firmly established itself in the field of hydro power generation. Up to now it has mainly been used to soft-start pump-turbines in pumped storage plants. In recent years, however, there has been a marked increase in the demand for continuous varspeed operation in pumping and generating applications. This paper presents a brief but comprehensive survey of static frequency converter based adjustable speed systems used in hydro power generation. The emphasis is put on the two systems built by ABB. The paper contains descriptions of the system concepts and includes explanations of the operating principles as well as information on the equipment employed. Finally we report on some topical projects.

1. INTRODUCTION

Variable speed in hydroelectric power generation was initially applied to pump-turbine units in pumped storage plants. In recent years, however, it has also been used in run-of-river plants. At present, pumped storage plants are the only proven energy storage facilities in widespread use. The potential of hydro varspeed becomes evident when we consider that more than 150 pumped storage plants, with a combined capacity exceeding 100'000 MW, are in operation worldwide. According to ASCE/EPRI sources (1989), in the U.S. alone 34 plants are already in operation with a further 33 projects proposed. There is a considerable demand both to equip new plants and to retrofit existing ones for varspeed. Until quite recently varspeed has been used only for the soft-starting and dynamic braking of reversible pump-turbine sets. Today, however, and especially when the flow of water is subject to change and/or if fluctuating head conditions prevail, varspeed can provide additional advantages. Owing to the removal of operational constraints, it is now possible to achieve better process control, improved performance, optimized water utilization and more cost effective exploitation of this naturally available energy source.

The fundamental problem with hydro varspeed is to achieve the energy exchange between the pump-turbine running at variable speed and the electric utility grid of fixed frequency. This paper deals with accepted solutions to this problem. It is limited to the electrical options based on Static Frequency Converters (SFC). The emphasis is put on features of the system and the SFC equipment.

---

1 Manager of Engineering Dept. UAE, ABB Drives AG, CH 5300 Turgi, Switzerland
2 Project Manager, Dept. KWHV, ABB Kraftwerke AG, CH 5242 Birr, Switzerland

When considering the implementation of variable speed based on SFC technology, three applications are of particular interest:

(a) Soft starting pump-turbine units
(b) Variable speed operation in pumping mode
(c) Variable speed operation in generating mode

(a) For base-loaded thermal units and especially with the proliferation of nuclear power stations, the problem of short-term energy storage has arisen. It is here that pumped storage comes into its own. It allows excess mains electricity, available when the demand is low, to be used to pump water to a high level reservoir. Subsequent peak loads are then met with hydro-electric power generated by operating the same motor pump in reverse direction, now as turbine generator, using the water from the high level reservoir as the energy source. The use of reversible pump-turbine units in pumped storage has created a requirement for rapid reversibility. Of the various run-up methods (direct, pony motor, etc. ...), the frequency run-up method, using an SFC together with the synchronous machine as the driving motor is best suited to this task.

(b) (c) Additional benefits can be derived from varspeed if power/waterflow adjustment is achieved by stepless control of the pump-turbine shaft speed. With fixed speed it is difficult to deal with fluctuating water head and to operate a pumped storage plant over a wide head range with consistent high performance. Continuous adjustment of the shaft speed will optimize the hydraulic/electric performance and thus substantially improve overall efficiency.

In the **pumping mode**, particularly in plants with extreme water head fluctuations, varspeed enables the pump to operate at near maximum efficiency or, if required, to optimize speed for best cavitation conditions. On the other hand, for a given head, varspeed permits the pump to be operated according to the amount of excess power available, thus taking maximum advantage of low cost electricity from the power utility system. Owing to the SFC's high speed active power response, the stability of the system is also improved.

In the **generating mode** also, varspeed provides better control over the generating process. The operational head range can be expanded around the design value without impairing the high efficiency. Furthermore, efficiency at partial load remains high and the stability of the system is also improved.

2. VARIABLE SPEED AC MOTOR-GENERATOR SYSTEMS

There are a variety of systems for operating AC motor-generator sets at variable speed. All are based on the combination of a suitable AC machine with one of the available static frequency converter (SFC) types. In hydro- electric power plant application two basic systems have become standard practice. The converter-fed synchronous machine (1) and the wound-rotor induction machine with sub/supersynchronous converter cascade (2). In both systems the prime function of the SFC is to exchange and control energy between an electric utility grid of fixed frequency and voltage and the motor-generator set operating at variable speed i.e. in general variable frequency and voltage.

2.1 Converter-fed synchronous machine (ABB MEGADRIVE-LCI)

"LCI-System" is an abbreviation, standing for Load Commutated Inverter system. As the concept and function of the system are well known, we shall restrict ourselves to a brief summary. Figure 1 shows a simplified diagram. The synchronous machine exchanges energy via the SFC and an isolating transformer with the AC power-system. The SFC is set up according to the DC link configuration of the current source type. Two identical thyristor converters in three-

phase bridge connection form the basis of the scheme. One is connected to the power-system. It is "line commutated" and operates at constant frequency. The other one connected to the machine, is "load commutated" and is operated with variable frequency. Thus the load has to supply the reactive power required for control and commutation of this converter. The overexcited synchronous machine fulfils this condition. The two converters are interconnected through the reactor in the DC link. It decouples the different frequencies of the converters. This type of SFC places practically no restrictions on the maximum design power. In pumping mode, when the machine operates as a motor, the line-side converter acts as a rectifier providing current control, the machine-side as an inverter commutating the current between the machine phases. The similarity to the converter-controlled DC machine becomes evident. Speed and torque (DC current) control can be handled in the same way as for the DC machine. The firing pulses for the machine-side converter are derived from the machine voltages (or alternatively from the rotor position) and are thereby in phase relation to the angular position of its rotor (self control mode). The machine is fully self controlled and hunting or loss of synchronism are ruled out. Generating mode can be achieved without including any extra equipment in the SFC. When the machine operates as a generator, the converter on the machine side acts as a rectifier, that on the line side as an inverter. In this way the flow of energy is reversed. Reversal of the machine's direction of rotation is also possible.

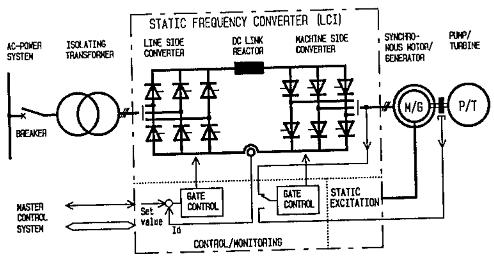


Figure 1 Converter-fed synchronous machine (system diagram)

A fundamental difficulty of the LCI system would appear to be that the machine cannot be self-started. The machine-side converter is load commutated, meaning it relies on the machine voltage for commutation. However, at standstill this is zero. A simple low cost solution, namely "DC pulsing mode", has become the standard starting method. In this mode of operation, which is applied in the frequency range from zero to several Hz, commutation is performed not by the motor voltage but is by line commutated pulsing of the DC link current.

The LCI-system is best suited to high and very high power, mainly single-motor applications and especially in conjunction with high speed requirements. ABB MEGADRIVE-LCI and -LCI.ST systems cover speed controlled drive and generator applications as well as starting equipment for the run-up and synchronization of synchronous machines. World-wide more than 350 units with a total power of about 1600 MW have been supplied, the majority of which have been commissioned and running successfully for years.

2.2 System with wound-rotor induction machine and cascade SFC of the cycloconverter-type (ABB MEGADRIVE-CASCADE.SS)

"CASCADE-SS" stands for CASCADE sub- and supersynchronous. Figure 2 shows a simplified diagram of the power part and controls, identifying the major components of the system, i.e. the doubly-fed wound-rotor induction machine, the three-phase cycloconverter for rotor supply and the electronic control system.

Each phase of the cycloconverter consists of a static converter setup with two antiparallel three-phase bridges, equipped with thyristors and corresponding line-transformers. These bridges are,

as a rule, operated in the noncirculating mode, meaning no circulating current is permitted between them. All converters are "line commutated" and are therefore independent of the load as far as commutation is concerned. Normal phase-controlled thyristors are used. The setup is the same as that used in the LCI-system. Like the reversing DC supply, the cyloconverter is also capable of four-quadrant operation, i.e. the cycloconverter output voltage and current can assume both polarities independently of each other. This property is required for active and reactive power control of the wound-rotor induction machine.

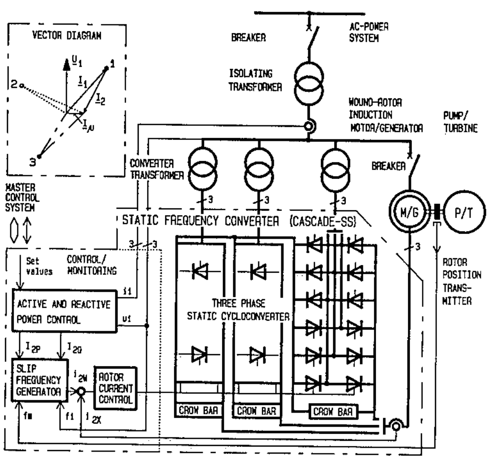


Figure 2 Wound-rotor induction machine with sub/super-synchronous converter cascade (system diagram)

The cycloconverter is operated in phase control. The firing angles of the thyristors are controlled so that (disregarding harmonics) a mean three-phase sine wave output voltage results. In addition, the control permits the adjustment of the frequency, the amplitude and the phase angle of the generated sine waves. So, the three-phase rotor current of the induction machine can be adjusted in frequency, amplitude and phase angle. Inversion of phase sequence poses no problem. Via the rotor current $\underline{I}_2$ (figure 2), the magnitude and phase angle of the stator current $\underline{I}_1$ and therefore the active and reactive power of the doubly-fed induction machine can be also kept under control and set to any required value in all four quadrants. The cycloconverter can transfer slip power in either direction, from as well as to the rotor. Thus, power control of the machine can be achieved in both the sub- and supersynchronous regions, independent of speed. For both regions and also for synchronous speed, where the excitation is by DC currents, pumping mode as well as generating modes are possible.

It is the task of the rotor current control to impress three-phase, sinusoidal, slip-frequency currents i_{2X} on the rotor via the cycloconverter. The set values i_{2W} for these currents are computed in the slip-frequency generator in such way that the motor-generator will exchange the required active and reactive power with the grid. To effect this computation the following quantities are used: The output signals I_{2P} and I_{2Q} of the active and reactive power control, the frequency f_1 of the grid and the rotational speed f_m of the motor-generator.

A thyristor crowbar limits the rotor voltages to the maximum permissible thyristor voltage in case of failures in the grid.

A total of 10 plants of the type described above have been built, most of which are in operation. The total installed power of these is 390 MW. One typical example is the frequency converter station "Seebach" which serves to control the exchange of energy between two frequency-elastic grid systems. This station, in which two 65 MW units are installed, has been in successful operation for 10 years.

3. VARSPEED INSTALLATIONS / PROJECTS

ABB's developments and first applications of high power AC drive technology date back to the mid-sixties. The pioneering achievements with the frequency start-up method by equipping the first gas turbine plants with SFC's of the LCI type in 1974 was followed in 1977 by the introduction of SFC's for the run-up of pump-turbine sets for pumped storage. Over the following 15 years more extensive use was made of adjustable speed AC drives in the electric power generation and industrial fields. Among other things, there was the realization of varspeed hydro pumping/generating. Today the ABB group can build on over 20 years experience with SFC drive systems, including applications with synchronous as well as induction machines. Known as the MEGADRIVE family, the systems cover the range from 1MW to 80MW and speeds up to 7500 rpm. In principle even higher ratings are attainable. The examples below represent selected equipment and highlight some realized varspeed hydro projects.

3.1 Soft-starting in pumped storage power plants

Static run-up equipment consists essentially of a static frequency converter (SFC) which draws power from the constant frequency/voltage utility grid and delivers it, in the form of variable frequency currents, to a synchronous machine in order to run it up to a certain speed. The LCI-system is best suited to this task.

Running-up and synchronizing a pump-turbine unit within several minutes is possible with SFC equipment designed with only a fraction of the machines power rating. Quick reversal of the unit (from either direction) can be achieved by regenerative braking into the grid via the SFC equipment. This method is particularly economical when the same SFC is used for the staggered run-up of several units.

Irrespective of its specific application, the SFC-based starting method meets the following general requirements: Negligible impact on the grid during start-up (soft start); practically no thermal or mechanical stresses on the machine; space-saving with freedom of installation; short erection and commissioning times; fully automated, easy to operate; high availability; high efficiency; almost maintenance-free; cost effective.

One topical example of an SFC run-up application is in the **Rocky Mountain pumped storage power plant**. This plant, built by Oglethorpe Power corporation and now under construction in north-west Georgia, not far from Rome, will have three single-stage reversible pump-turbine units [1]. Each synchronous generator-motor is rated at 283 MW as a generator respectively 287 MW as a motor, at a voltage of 20 kV. The rated speed at 60 Hz is 225 rpm. One SFC is provided for the staggered run-up of all three units. Completion of the project and the commencement of commercial operations are scheduled for 1995.

The simplified single-line diagram in figure 3 gives an overview of the SFC run-up application. The SFC is connected between the power system and the starting bus. From here it can be selected to operate on any of the motor-generator sets. Apart from some additional circuit breakers for switching the SFC in and out, for selecting the machine and for switching between the redundant supplies, all that is needed besides the SFC is an isolating transformer and a matching transformer for adapting the SFC output to the motor-generator. When starting from standstill and running at up to 10 % speed, i.e. at very low frequencies, this transformer is bypassed with isolators (IS). For the run-up the static excitation units (EXC) of the motor-generators are used.

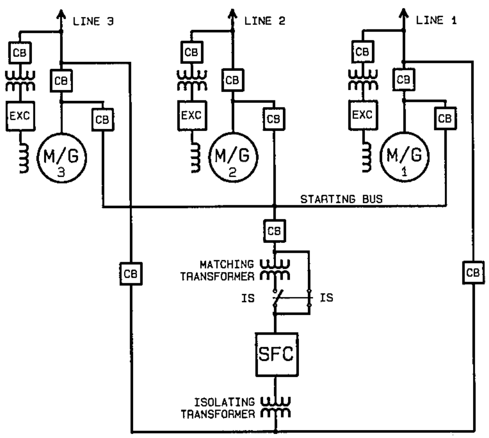


Figure 3 Rocky Mountain SFC softstart application (single-line diagram)

The SFC is designed according to the required run-up time and the load torque characteristic and inertia prevailing during run-up. The power at the starting bus needed to accelerate a dewatered pump-turbine from standstill up to 225 rpm within 8 minutes is 16.5 MW (at 225 rpm). For this application a compact, watercooled SFC of the MEGADRIVE-LCI.ST type, as shown in figure 6, has been selected. It is a mono-block, metal-clad construction of modular design. All modules, power packs as well as all control cubicles, are mounted on a common base frame. The power pack is controlled by a powerful microprocessor based controller. This controller also monitors the automatic sequencing of all modes of operation.

In the Rocky Mountain project any unit can be selected for any of the following modes:

- run-up and synchronization for pumping
- back-to-back start and synchronization for pumping
- run-up and synchronization for synchronous condenser operation
- dynamic braking from pump and also turbine operation for stopping a unit
- rotor positioning (with adjustable torque)

3.2 The PANJIAKOU project

In 1986 the P.R. of China undertook the upgrading of their Panjiakou hydro power plant, situated roughly 200 km east of Beijing. At that time the plant consisted of only one 150 MVA conventional generating unit. The electrical equipment, manufactured by ABB to upgrade the plant for pumped storage, comprises three synchronous-type motor-generators, each rated at 98 MVA at 50 Hz/125 rpm (generator mode) and one 63 MW static frequency converter (SFC). The motor-generators were designed for two synchronous speeds (125 and 142.86 rpm by pole-switching). The single-stage Francis reversible pump-turbines were supplied by Sulzer-Escher Wyss [2]. The pumped-storage extension was put into operation in December 1992.

Apart from providing additional generating capacity, especially for peak load, the upgrade project was carried out for two main reasons. Firstly to improve the flexibility of the electric grid, i.e. to better balance electricity supply and demand. Secondly to gain experience with varspeed pump-turbine operation and optimize gross efficiency over the wide water head range (about 35 to 85 m). The single line diagram in figure 4 shows the electric system of the pumped-storage extension. The SFC can be used optionally with any of the three motor-generators (M/G), be it for start-up only or for continuous varspeed operation. The SFC is an LCI-type as described in section 2.1 and is rated at 63 MW / 13,8 kV for the varspeed pumping mode at 45,5 Hz. For running up a unit to 50 Hz and synchronizing it, a fraction of this power will suffice. The SFC's power section is

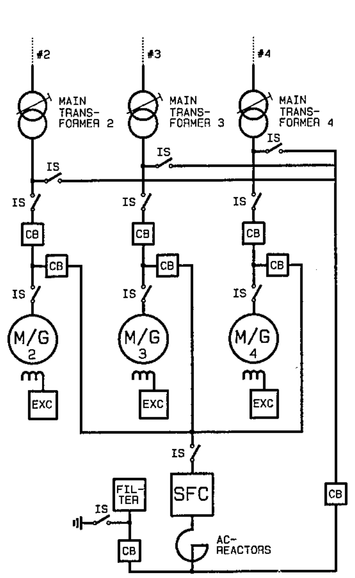


Figure 4 Panjiakou varspeed application (single-line diagram)

watercooled and of fuseless design throughout. The voltage is matched to the M/G by means of 14 series-connected thyristors per bridge leg. One of them is redundant, i.e. if one in each leg were to fail the SFC would still operate at full capacity without any break. Triggering and monitoring of all thyristors is by light signals via fibre optics. Firing energy is derived from the power circuits. The SFC is arranged for 6-pulse operation. It produces current harmonics that in principle could interfere with the M/G and the utility grid. The resulting additional losses and pulsating torques have been given careful consideration in the M/G design. Filters tuned to the lowest harmonics (5th, 7th, 11th and 13th) are installed to reduce the impact on the grid. In addition they provide reactive power to correct the lagging power factor of the SFC. A powerful microprocessor-based programmable controller performs all SFC control and monitoring tasks and also supports diagnostics. The SFC is of modular design. It is split up into several physically separate assemblies for easy transport and uncomplicated erection. The largely standardized modules comprise all power packs, the control cubicles and the water cooling unit. The modular concept and the programmability of the controls provide the flexibility often required to comply with specified design criteria of customers.

The following SFC-operation modes are implemented in the Panjiakou application:

- run-up with watered pump (valve closed) and synchronization for pump operation
- run-up with dewatered pump and synchronization for synchronous condenser operation
- varspeed pump operation with SFC
- varspeed turbine operation with SFC
- dynamic braking of dewatered turbine

The extension for pumped storage, i.e. the SFC and the three motor-generator units, has been commissioned and is in operation. All modes of operation with the SFC have been successfully tested and no major system problems have occurred. System tests using varspeed up to full SFC power are still in process.

3.3 The COMPUERTO project

One example of a CASCADE-SS application is currently being realized in Spain. Iberdrola, Spain's largest hydro electricity company, is working on a retrofit project to convert its conventional power plant at Compuerto to varspeed. This is a pilot project to gain experience with varspeed in order to evaluate its potential for use in further plants with higher ratings. The Compuerto station functions only as a generator, is used for agricultural irrigation and only operates in the summer. ABB-Switzerland and ABB-Spain have been given the contract to modify one of the two 10MW synchronous generators (figure 5) from fixed to variable speed. This was done in the winter of 1992/93 by replacing the rotor of the synchronous machine with a three-phase wound-rotor, thus converting the synchronous machine to a doubly-fed induction machine. This new rotor is supplied from a cycloconverter. As a result of this modification the generator can be operated with variable

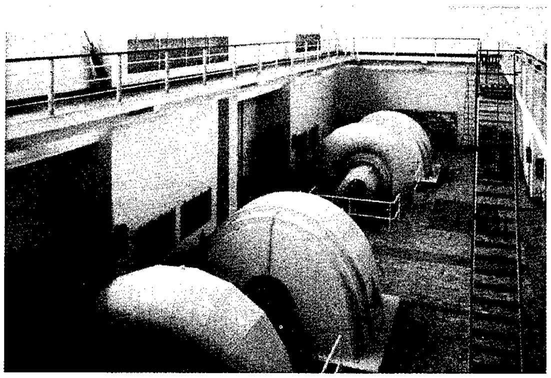


Figure 5 Machine house of the Compuerto hydro power station with two turbine generator sets, 10 MW

speed in a range of 10% around synchronism (600 rpm) as described in section 2.2. For this application a compact, watercooled and standardized SFC was selected. The three-phase cycloconverter is rated at 700 V, 1800 A, 0 ... 5,3 Hz. The power section and control cubicles are mounted on a common base frame, allowing easy testing, erection and commissioning. All control functions for the cycloconverter and its application to the generator, as well as for control of the turbine governor, are implemented in a powerful programmable high speed controller (PSR).

The following operation modes can be selected:

- run-up to the speed control range via turbine
- synchronisation, independent of speed, via the machine power control system
- varspeed turbine operation with cycloconverter

After commissioning in May/June 1993 it is the aim of both Iberdrola and ABB to gain experience as regards the following:

- optimizing the functions of the electronic control system of the wound-rotor induction machine, when working together with the turbine governor (this is important both in normal operation and in the event of failures in the network and/or in the hydraulic system)
- the behaviour of the equipment, especially the newly retrofitted induction machine, in this new application
- the anticipated economical advantages

4. SFC-EQUIPMENT

The static frequency converter configurations for the various drive systems are well established. In today's varspeed hydro application, the basic converter circuit topology is the traditional, well proven three-phase bridge circuit as used in many drive applications and also in HVDC stations with power ratings up to the gigawatt range. To cope with these very high voltages a large number of thyristors can be connected in series without any problems arising. Use of the most up-to-date power electronics technology, based on both progressive macroelectronic components (more powerful thyristors) and microelectronic components (micro-processors with increased processing speed and high density memory chips) has led to better, more cost-effective SFC's.

The SFC equipment offered by ABB for varspeed hydro applications is largely standardized and based on modular mechanical and electrical designs. Because of their modularity these SFC's are flexible enough to cope with the various application-specific demands made upon them. In most cases all modules and components are housed in metal, block-mounted cabinets, on a

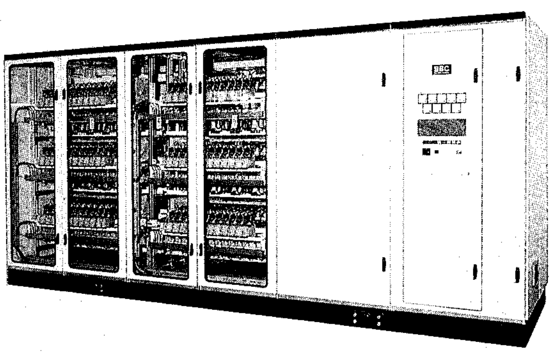


Figure 6 Typical water-cooled LCI-type block-mounted SFC (10 MW, 0 ... 100 Hz)

common base frame forming a single unit for transportation (figures 6 and 7). The use of this type of equipment considerably simplifies and shortens factory testing, erection, commissioning and servicing. In applications for variable speed pumping and generating operation, very high converter power with high voltages is needed. For voltages of up to 20 kV as many as 18 thyristors per leg may have to be series-connected and assembled in vertical columns. As a rule, these large converters must be split up into several units for transportation.There is the choice of either air or water cooling. **Figure 6** shows a typical **water-cooled LCI-type block-mounted SFC** (10 MW, 0 ... 100 Hz). The power section on the left side is of fuseless design throughout. Both rectifier and inverter modules are housed in two cubicles in pairs. These modules comprise the thyristor columns, snubber circuits, firing circuits and measuring transformers. Thyristors and heat sinks are sandwiched in horizontally arranged columns. Deionized cooling water is pumped in closed loop through the heat sinks and snubber equipment. The water cooling module comprising primarily deionizer, pump and water/water heat exchanger, is also mounted on the block. Thyristor gate-triggering is via pulse transformers (indirect light triggering is used with high converter voltages). The control module (figure 7 right) comprises all equipment to control the SFC. It is microprocessor based. All control functions: regulation, automatic sequence control, moitoring, diagnostics and protective back up functions etc. are implemented in a single powerful, programmable high speed controller (PSR) which has been especially designed for power electronic applications. Interfaces for communication with the PSR are available. The digital design of the control enables a high degree of flexibility and allows easy change of control functions so that various requirements can be satisfied. Due to the significant reduction in the number of electrical components in the digital control, a very high level of reliability and availability has been attained.

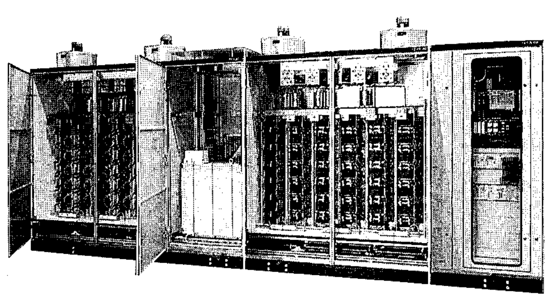


Figure 7 Typical air-cooled LCI-type block-mounted SFC (24 MW, 0 ... 52 Hz)

Figure 7 shows an **air-cooled LCI-type SFC** (24 MW, 0 ... 52 Hz) for soft-start application in a pumped storage plant. It is also a block-mounted design similar to the one in figure 6. Here, however, the DC link reactor (iron cored) is also mounted on the block between the rectifier and the inverter modules. Thyristors and heat sinks are assembled in vertically arranged stacks. In this

design the air is drawn through the thyristor stacks and the cubicles by fans fitted on top of the block. With this standardized, block mounted SFC equipment erection on site, connection of all cables/buses and commissioning can be carried out with ease.

5. CONCLUSION

With the introduction of variable speed, based on static frequency converter technology, hydropower generation and especially pumped storage have entered a new phase. Where severe constraints prevail there are considerable benefits to be gained both in plant efficiency and process control. Mature SFC technology and the use of proven equipment also combine to make varspeed attractive for hydro power generation. Applications for the soft-start of pump-turbines have become standard practice and the application of variable speed to pumping and generating is attracting growing interest.

Based on many years of experience in manufacturing electric machinery and power electronics equipment, ABB has developed two different systems for these applications: One for synchronous machines and one based on the induction machine. The static frequency converter equipment used in these systems is compact and largely standardized and is based on modular mechanical and electrical designs. The control functions are implemented in a programmable controller. This equipment has the intrinsic flexibility to meet customer-specific requirements and facilitate quick erection and commissioning. ABB has the capability to undertake the engineering of the complete electrical system.

All field experience and results obtained so far have strengthened our belief that varspeed technology will play an increasingly important role in the future of pumped storage, both in new and retrofit projects.

REFERENCES

[1] D.R. Murphy et al, "Rocky Mountain Project-Overview", Waterpower '91, p. 1636

[2] M. Jaquet, "The Panjiakou Pumped Storage Station, Part I: Hydraulic Equipment" Fourth ASME International Hydro Power Fluid Machinery Symposium, December 1986

SUMMIT PUMPED STORAGE PROJECT DISTRIBUTED CONTROL SYSTEM

by Øistein Andresen[1] and Tore Jensen [2]

Abstract
The paper presents a Distributed Control System (DCS) which is being considered for the Summit Pumped Storage Project. The paper discusses the control functions to be performed by the DCS during the different modes of operation, and how the DCS interfaces with the rest of the plant control equipment to form a fully integrated state-of-the-art control system for safe start-up, shutdown and continuous operation of the hydroelectric units and associated ancillary equipment.

The paper discusses the system configuration, and how this configuration with distributed architecture and independent functional units minimizes the impact of a single system failure. The paper also discusses how redundancy will be applied on critical levels to further improve the system availability.

Introduction

The Summit Pumped Storage Hydroelectric Project in Norton, Ohio has been licensed by the Federal Energy Regulatory Commission (FERC). The project is a 1500 MW hydroelectric pumped storage facility scheduled for completion in the year 2000. The facility will make use of an existing limestone mine as its lower reservoir. A new impoundment to be constructed at ground surface will serve as the upper reservoir. An underground powerhouse will contain six 250 MW pump-turbine/motor-generator units.

---

1 Area Manager, ABB Energi AS, P.O.Box 214 Økern, N-0510 Oslo, Norway
2 Section Manager, ABB Energi AS, P.O.Box 214 Økern, N-0510 Oslo, Norway

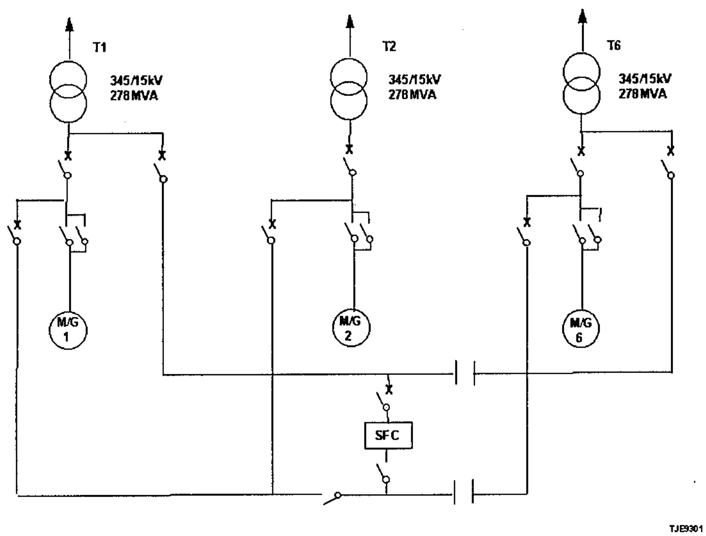


Figure 1 – One-line diagram

A fully integrated microprocessor based Distributed Control System (DCS) will be provided for the control and monitoring of the six pump-turbine/motor-generator units, their respective auxiliary equipment, and the station common equipment.

The Summit project is still in the design phase and therefore details regarding plant operation and control system design have not been finalized. While the DCS described in this paper is currently under consideration, the control system design has not been finalized.

Functional requirements
The principal functions of the hydroelectric units will be to generate electric power when operated in the generation mode, and to pump water from the lower to the upper reservoir when operated in the pumping mode.

Operation in the Generating Mode. In the generating mode, the units will be capable of operating at variable output and therefore will be suitable for system load following and load regulation, including the ability to accommodate rapid changes in output in order to meet varying system conditions.

Operation in the Pumping Mode. Variation in unit discharge in the pumping mode is not practicable with single stage reversible pump turbines; therefore, the units will operate at a fixed motor input. In the pumping mode, the station discharge and hence required pump input power will vary with changes in reservoir water levels.

In addition to pumping and generation, other modes of operation will be possible with the units as follows:

Spinning Reserve (turbining). The runner area unwatered with compressed air, the penstock valve closed (spiral case may or may not be watered), and the unit motoring at synchronous speed in turbine direction;

Spinning Reserve (pumping). The runner area unwatered with compressed air, the penstock valve closed, and the unit motoring at synchronous speed in pumping direction;

Synchronous Condenser (turbining). Unit in spinning reserve in turbining direction and used for reactive compensation;

Synchronous Condenser (pumping). Unit in spinning reserve in pumping direction and used for reactive compensation;

Back-to-Back Motoring. The runner area unwatered with compressed air, the penstock valve closed, and the unit motoring in pumping direction. The motor-generator terminals are connected to the motor-generator terminals of another unit generating in turbining direction; and

Speed-no-Load. Unit operating in turbine direction at synchronous speed with no load on the units.

Since the plant will be used for system load following and load regulation, and will operate in various modes, the controllability of the plant, and hence the performance of the control system is essential. Frequent start/stops and mode changes where change-over times sometimes are crucial to meet varying system conditions requires a fully automatic control system. The DCS will therefore be designed to accommodate fully automatic start and mode transfer with a minimum of operator interaction. Figure 2 illustrates the automatic start and transfer capabilities of the DCS.

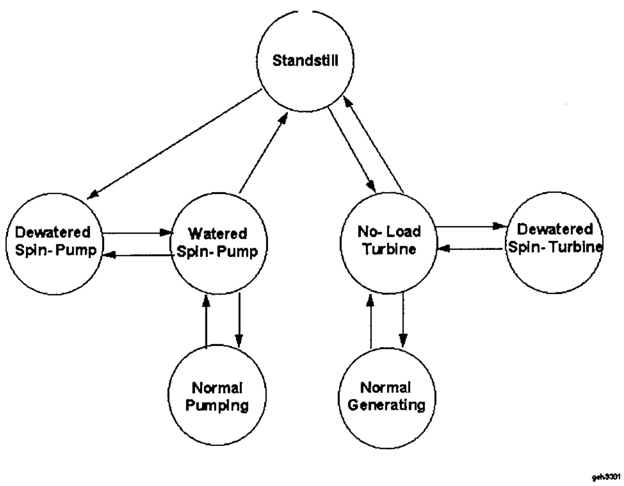


Figure 2 – DCS automatic start and transfer capability

When a unit is started in the turbine direction, whether it be for operation in the generating, spinning reserve or synchronous condenser mode, it is started as a conventional hydro unit by water discharged through the turbine. In any of these modes the DCS will be capable of bringing the units from a dead stop to the selected operating mode, or from one operating mode to another, upon a single key stroke or operation of a single pushbutton.

Starting of a unit in pumping direction is accomplished by means of one of the two Static Frequency Converters (SFC) furnished for the plant. During a start the DCS will automatically position the disconnect switches and the circuit breakers in the starting bus to connect the unit to the selected SFC. Following this, the DCS will actuate the SFC which smoothly accelerates the unit to synchronous speed by providing variable frequency power to the motor–generator terminals.

Starting of a unit in pumping direction is also possible by means of another unit in back–to back connection. During a back–to–back start the DCS will automatically position the disconnect switches and the circuit breakers in the starting bus to connect the motor–generator terminals of the two units, and control the run–up of both units to synchronous speed. Once the pump unit is synchronized and connected to the grid the other unit will be automatically shut down.

In any mode of operation the DCS will upon operator request or detection of any condition that is detrimental to further operation, provide safe shutdown of the unit.

All start and stop sequences will be monitored to ensure completion. Alarms and indication of incomplete sequence failure and incomplete sequence cause will be reported to the operator workstations. Furthermore, the operator workstations will be provided with start and stop sequence displays which will provide the operator with the capability to observe unit start and stop sequences step–by–step.

The operator workstations will be the primary man–machine interface. Start/stop and mode selections will be accomplished by means of function keys on the keyboards and display menus on the Video Display Units. It will also be possible to perform start/stop and mode selections by means of push–buttons in the control panels.

In addition to start/stop sequencing the DCS will provide control and monitoring functions, such as:

* Automatic adjustment of MVAr and MW output of a single unit or the entire plant to the set point entered by the plant operator or the load dispatcher;
* Monitoring of water level in upper and lower reservoir;
* Sequence of event recording with two milliseconds system wide resolution;
* Alarm annunciation. Alarm lists may be viewed from the operator workstation unit by unit or for the entire plant;
* Temperature monitoring (RTD inputs);
* Trend data logging;
* Automatic disturbance trend recording (pre/post trip logs);
* Automatic transfer of the station service system from the normal source of supply to the standby source upon loss of power;

System configuration
The DCS consists of distributed process controllers and operator workstations interconnected in a network via high speed data communication links. The process controllers perform independent process control and process interface, and the operator workstations provide the man machine interface. To minimize the impact of a process controller failure, each unit will be provided with a separate process controller functionally independent of the others.

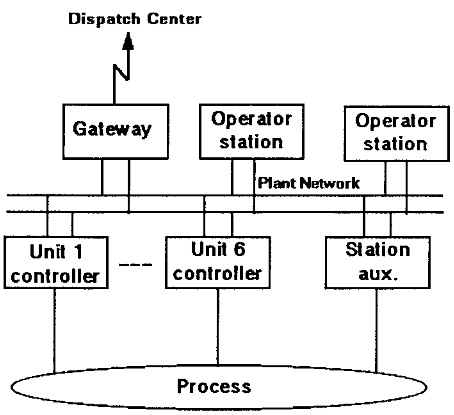


Figure 3 – System configuration.

Distributed process controllers
The process controllers will monitor the status of designated inputs, direct the status of designated output functions and sequentially start–up or shutdown the hydroelectric plant according to the system operational requirements.The real–time process database is entirely distributed to the process controllers. These process controllers are functionally complete and able to perform logic control, regulatory control and advanced calculations. This ensures that all processing and control can be performed on the spot; i.e. data does not have to be sent elsewhere for processing and the results returned. Consequently, the control system can be structured according to the process requirements without restrictions imposed by limitations in the control system. The result is a "clean" structure with a high degree of integrity.

The process controller system software comprises a real–time operating system. Application programs are executed cyclically on defined priority levels. Cycle times will normally be selectable between 10 ms and 2 sec. Several supervisory functions automatically monitor system operation, and report and indicate any detected fault. Faults are indicated with LEDs on relevant boards, and by system error messages reported to any operator workstation.

Process interfacing is through local or distributed input/output (I/O) modules. Distributed I/O modules are connected to the process controllers via a field bus system. The field bus system will also be used for serial interface with the static frequency converters, static excitation systems and turbine governors.

Operator workstation

The operator communicates with the system and the process through colour–graphic video display units, keyboards and printers. A trackball or a mouse is used to point and select the display to be called up; process device to be controlled; report to be printed; etc., resulting in convenience and intuitive operation in the process control.

The use of open technology and standards, both industry and international, ensure useful openness. Examples of important standards are: SQL – a standard for accessing information from databases; X Window System – a protocol allowing users to interface with applications in other computers from windows in their own; and OSF/Motif – a standard defining how window applications are to interact with users.

The operator interface functionality includes advanced features for communication with the user: presentation of user–defined process displays, standard displays, trend displays, reports and various lists, effective operator dialogue for manual control, self–diagnostics, and display of system status.

Operator workstations are connected directly to the plant network. In addition, the TCP/IP protocol can be used to provide interconnection of operator workstations for purposes which are not so time–critical and to communicate with external computers. Utilizing the X Window System, windows can be opened on the operator workstation screen for working with other systems through the TCP/IP network.

The man–machine interface system will respond to any operator request or change in the process in a reasonable time. During normal operation the system will meet the following: display exchange time shall not exceed 3 seconds; the time from when a digital process input changes state to when it is updated on the screen shall not exceed 1 second; time from operator command to process output shall not exceed 1 second; time from operator command to update on display shall not exceed 3 seconds; analog values shall be updated at least every 9 seconds.

Plant network
The plant network provides real–time communication between operator workstations and process controllers. The plant network is also used for cross communication between process controllers, and for communication with the dispatch center via gateway.

The plant network is based on the IEEE 802.3 standard. The same physical cable can be used for the different higher–level protocols that are based on the IEEE standard. The plant network is used for the time critical, real–time communication within the control system. In order to ensure integrity for each plant network, it is possible to limit the access to information, and the amount of resources that can be occupied from outside the network.

The communication system is self configuring. This means that the user only has to connect the stations together physically, while the system itself maps out the topology of the network and establish the required communication paths through it. This is particularly important for large systems and when expanding existing installations.

Redundancy
Extremely high availability is required from the control system. To meet such requirements, critical parts are duplicated, In process controllers, the central unit (CPU and memory) are duplicated. While the active unit is running the process control functions, the backup unit is in "hot standby", i.e. continuously updated by means of an automatic check pointing procedure. Takeover takes less than 25 ms, and is totally bumpless. Communication units and power supplies are duplicated. The plant network is equipped with dually redundant links and alternative–path selection to maintain communication at all times, even if individual links drop out. Operator workstations are entirely duplicated .

Communication with dispatch center
The DCS will have the capability of interfacing a dispatch center (DC) for remote operation of the plant. The interface will be designed to permit 2–

way serial data exchange between the two control systems. The interface protocol will conform to IEC TC57. The data integrity is high for the data transmission (Hamming distance 4). This is achieved partly by horizontal and vertical parity check for each transmission.

The dispatch center sends cyclic data request telegrams to the DCS system. An output data telegram will temporarily interrupt the polling. Data may be given different priorities so that the most important data is reported first. Single indications are normally used for alarms. Dual indications are normally used for breaker and isolator positions. This indicating method increases the security and permits indication of mid–position. Inputs which are oscillating because of faults in the local equipment may be automatically blocked at optional frequency.

The measured values are read at optional cycle time. Upon reading, the present value is compared with the last reported value. If the difference is greater than an optional deadband, the value is reported to the DC. With this method a short update time is obtained compared with periodic reporting, independent of changes. Every analog measured value may be supervised against 4 limits, optional for each object. The limits have optional hysteresis.

The DCS may accumulate pulses from external pulse transmitters (e.g. energy measurement). Reading of the accumulated value is done both at end–of–period and intermediate period.

The command sequence always comprises selection of an object, and execution. A command may optionally be done as an immediate execute command or as a select before execute command. The select before execute command offers an increased security against faulty maneuvers.

Conclusion

The Summit Pumped Storage Project with its comprehensive operating scheme will greatly benefit from a state–of–art distributed control system. The DCS provides automatic control and monitoring capabilities which will enable the operator to quickly start and stop units and change operating modes to respond to varying system conditions. The DCS will therefore be an important tool in maximizing the dynamic benefits from this project. The distributed architecture and the use of independent functional units minimizes the impact of a single system failure. Redundancy applied on critical levels further improves the overall system availability.

Spring-Type Thrust Bearings Can Be Monitored

Michael E. Coates [1]

Abstract

New England Power Company owns and operates 45 hydroelectric turbine/generators at 15 stations on the Connecticut and Deerfield Rivers. Most of these turbine/generators are vertical units with a variety of types of thrust bearings. The generating stations have experienced several problems with thrust bearings throughout their history but the most difficult to solve have been in thrust bearings with a tilting-pad, spring-type design.

As a part of a recent contract for a hydroelectric turbine/generator, a combined New England Power Company and New England Power Service Company project team specified that the generator maker design its spring-type thrust bearing to accept a permanent monitoring system to measure both pad displacement and load in all operating and non-operating conditions. The project team engineers have selected each component of the system on the basis of the maker's bearing design information and calculated loads.

The project team will closely follow the bearing and monitoring system manufacture, testing, and installation. Our intent is to collect baseline data during unit erection, bearing runs, testing, startup, and load rejections. Our long-term plan is to extract and analyze thrust bearing data immediately before and after unit inspections.

----

[1] Senior Mechanical Engineer, New England Power Service Company, 25 Research Drive, Westborough, MA 10582

Background

In recent years, New England Power has had several hydroelectric generating units experience problems which appeared to be related to thrust. The problems have been frustrating to solve, primarily because of difficulty in determining if the thrust bearings were performing as they should. Some examples are:

- In one spring-type thrust bearing, in response to New England Power's frequent need to dress the babbitt on the thrust bearing pads, the turbine manufacturer recommended design improvements which would decrease hydraulic thrust. These improvements were implemented during the unit's ten-year overhaul. After the overhaul, the thrust bearing pads began to show what appeared to be overheating staining, indicating an increase rather than a decrease in thrust. The turbine manufacturer and the generator manufacturer were unable to demonstrate, other than by calculation, how the bearing's thrust might have been changed during the overhaul. The calculations predicted results opposite to actual observations.

- In the same thrust bearing, several years after the overhaul, some of the bolts which keep the springs compressed began to fail. The broken bolts were discovered during an annual inspection. During investigation, bolts continued to fail. The only way to detect bolt failure was to take the generating unit out of service long enough to drain the bearing reservoir, jack the rotating parts, remove each bearing pad, and inspect each bed of springs. A consultant found that the failure mechanism was fatigue cracking but no thrust or operation changes in the bearing's 17 year history seemed to increase cyclic loading.

- One unit's thrust bearing was completely disassembled and cleaned after the wipe of a guide bearing immediately above the thrust bearing. The pad and springs were removed, cleaned, and replaced. No damage was apparent to the thrust bearing. Upon reassembly and startup, the thrust bearing immediately wiped. After rebabbitting and wiping the bearing again, a very painstaking process was performed to avoid another bearing wipe. The bearing was assembled, the rotating parts were turned ten full revolutions, the pad was removed, the babbitt high spots were hand scraped, and the pad was reinstalled. This process was repeated until the thrust bearing runner was contacting more than 80% of the pad's surface. Visual inspection had indicated that the bearing was fit for service after the initial cleaning and reassembly. No means were

available to determine if the bearing was performing properly prior to returning the unit to service.

New England Power Company's maintenance personnel and New England Power Service Company's plant support engineers have become suspicious of spring-type thrust bearings. The problems' root causes have been elusive and we have not been able to develop a temporary or permanent retrofit system to instrument a spring-type bearing with any reasonable simplicity or cost. The engineers have concluded that there is nothing inherently wrong with the capabilities of this type of bearing, but that problems related to thrust in spring-type bearings are nearly impossible to troubleshoot.

In 1991 New England Power solicited proposals for two new turbine/generators. The preferred vendor offered a spring-type thrust bearing. Prices for alternate bearings were prohibitive. During negotiations, the project team decided to accept the spring-type bearing on the condition that a load and pad-position monitoring system be designed as an integral part of the bearing.

Thrust Bearing Purpose and Function

The purpose of a thrust bearing is to prevent lengthwise, or axial, motion of a rotating shaft and its attached rotating components. The bearing transfers thrust from the rotating parts of the machine to the stationary parts of the machine and eventually to the supporting stationary structures.

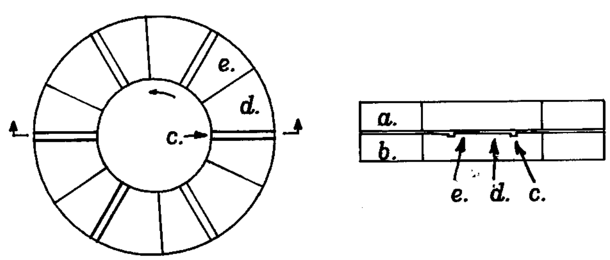
Vertical shaft machines require that thrust bearings carry the weight of the rotating parts and the thrust of the water acting on the turbine. The weight of the rotating parts is carried by the thrust bearing as dead weight whenever the machine is assembled, regardless of operating condition. The thrust of the water acting on the turbine varies depending on the volume of water passing through the turbine. The thrust varies approximately with wicket gate position. If the turbine wheel is oriented so that flow is downward, the dead weight of the machine and the hydraulic thrust of the water both act downward and are additive.

The bearing depends on a thin film of oil to support the thrust of the machine and keep the bearing surfaces from contacting each other. Most vertical shaft thrust bearings form the oil film by the action of the rotating part of the bearing, thus it is a hydrodynamic film. The rotating part draws a thin wedge of oil into the gap between the bearing surfaces. The oil wedge must be the correct thickness to provide the desired pressure distribution, supply a flow of oil across the entire pad surface, and ensure that the oil

thickness never falls below the minimum needed to keep the bearing surfaces from contacting. The wedge shape is formed by the geometry of the two bearing surfaces. A slight taper between the bearing surfaces in the direction of rotation is very effective in forming an oil wedge and film. The amount of separation required to form a wedge is very slight.

Generic Types of Thrust Bearings

Thrust bearings for vertical machines usually have two major parts: a running surface which is a continuous face and a set of segmented stationary pads. The segmented pads fall into two categories, fixed-pad and tilting-pad. Fixed-pad bearings most commonly are constructed with a tapered land at the leading edge of the pad. The tapered portion of the pad allows oil to be drawn into the bearing clearance and form the hydrodynamic oil wedge. Figure 1 shows a simplified sketch of a tapered-land bearing.

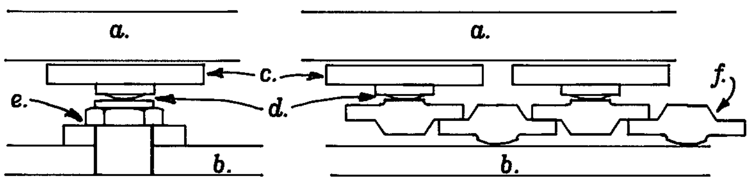


Plan *Section*

(a) Bearing Runner, (b) Base Ring, (c) Radial Slots, (d) Tapered Land, (e) Flat Land

Figure 1 - Tapered Land Bearing

Tilting-pad bearings are made in three types, jack-screw, self-equalizing and flexible-support. It is the tilting action of the pad which encourages formation of an oil wedge. Jack-screw and self-equalizing bearings normally have pads which are constructed with a pivot or rocker on the underside of the pad. This pivot or rocker allows the pad to tilt under the pressure of the oil film to form an oil wedge. In conventional turbines, the pivot or rocker is often located beyond the center of gravity of the pad in the direction of rotation, so that the pad tilts slightly into rotation. Figure 2 shows sketches of jack-screw and self-equalizing tilting-pad bearings.



(a) Bearing Runner, (b) Base Ring, (c) Pad, (d) Rocker or Pivot, (e) Jack Screw, (f) Equalizing Links

Figure 2 - Tilting Pad Bearing Sections

Flexible-support tilting-pad bearings use a system of springs, elastomeric cushions, oil pressure cylinders, or flexible metallic supports under each pad to permit the pads to tilt to form an oil wedge.

The generic types of bearings differ, not only in how they form a hydrodynamic oil film, but also in how they distribute the load on the bearing. Jack-screw bearings have a large screw under each pad which is adjusted manually to distribute the dead weight of the machine equally between all pads. Self-equalizing bearings have an interconnected mechanical linkage under the pads which tends to distribute the vertical thrust equally between all pads. Neither fixed-pad bearings nor flexible-support bearings have a means of adjusting thrust on the pads once the bearing is manufactured. The thrust distribution among the pads, then, becomes a function of design and fabrication tolerances. The fixed-pad bearing has the potential to be constructed to tight tolerances since the pads can be attached to their support ring and machined as a unit. It follows that a bearing made to very tight tolerances would tend to have relatively equal thrust distribution between the pads. Flexible-support bearings are made of many parts which must be made to operate independently of one another. The parts must be fabricated independently, each with its own tolerance.

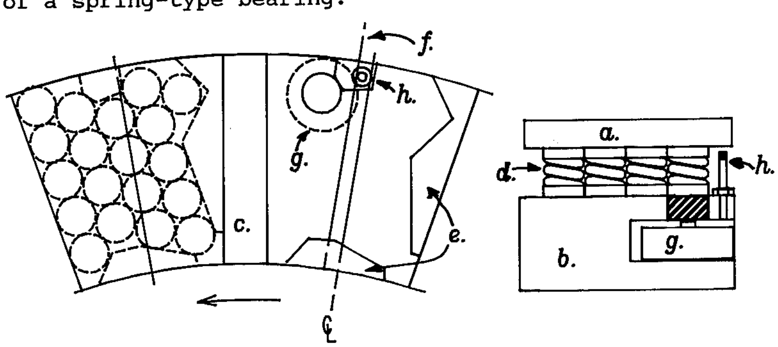
Flexible-support bearings are the most susceptible of all types of vertical shaft thrust bearings to experience unequal load distribution between pads. This susceptibility is due to the independence of the segmented pads, the lack of adjustability after manufacture, each pad's many independent parts, and the the sum of the parts' individual tolerances.

Each bearing has many other accessories which contribute to proper operation but which are not directly involved in

transfering thrust. One auxiliary system which is related to thrust, and which many bearings have, is an oil lift system. An oil lift system is used to provide lubricating oil between the bearing surfaces at times when the bearing cannot produce its own hydrodynamic film. When the rotating parts are at a standstill, just starting, or stopping, bearings often will not provide a sufficient oil wedge and film. An oil lift system pumps oil from the bearing reservoir and injects it between the bearing pads and runner. The pressure of the oil lift system must be sufficient to separate the rotating and stationary parts of the bearing and create a cushion of oil across the pad surface.

Design of Spring-Type Bearings

The spring-type (or GE-type) bearing is designed so that the segmented pads function independently of each other. The pads are separated by radial slots as they are in other types of bearings. The surface area of the bearing is selected to keep the maximum pressure on the surface of each pad below some critical value. Each pad rests on a bed of springs. The number of springs under each pad is designed to allow the pad to displace under load but not exceed certain displacement limits. The springs are arranged under the pad to allow the pad to tilt under hydrodynamic pressure. The tilting axis is commonly parallel to and just beyond, in the direction of rotation, a radial line through the center of gravity of the pad. Figure 3, items (a) through (f), show the common components of a spring-type bearing.



Partial Plan *Tilting Axis Section*

(a) Pad, (b) Base Ring, (c) Radial Slot, (d) Spring Assembly, (e) Spring Arrangement Spacers, (f) Tilting Axis, (g) Load Cell, (h) Proximity Probe

Figure 3 - Spring-Type Bearing

The springs are made from heavy stock and shaped in a closely coiled cylindrical helical pattern with ground open ends. Since the spring has ground open ends, its initial compression under load is not linear. The springs are individually assembled with end clamps and bolts. Each spring assembly is compressed with a hydraulic press under a force equal to the rotating parts dead weight supported by that spring. The bolt is tightened so that the end clamps keep the spring compressed to this pre-load length. Each spring assembly is precision machined to the same length. Pre-loading the spring compresses the spring beyond the non-linear portion of compression and ensures that when the springs begin to compress under live load, the spring constant is linear throughout its action.

Below the bed of springs is a solid base ring which is designed to transfer the thrust, with minimum distortion, from the several beds of springs to stationary structures. The bearing rotating and stationary surfaces operate continuously immersed in a bath of oil.

Monitoring the Bearing

Many types of bearings have designs which are relatively straightforward to troubleshoot. For example, in self-equalizing bearings, application of a strain gage to one load-carrying part of the bearing yields information on thrust in all parts of the bearing. This information is useful and accurate whether the unit is operating or not. If someone suspects that a generating unit is experiencing a problem due to a change in thrust or a change in the bearing, strain gage information will answer many questions.

A spring-type bearing is not so simple to instrument or analyze. This is because of several design features.

- Each bearing pad rests on its own bed of springs and is not linked to the other pads. Also, due to the many parts and the sum of the manufacturing tolerances, the load may not be equalized between the pads. So for complete and accurate information from a spring-type bearing, each pad would have to be instrumented.

- Each spring is pre-loaded and clamped at a length equal to the compression of the unit's dead weight. So the bearing, with all pads sitting on beds of pre-loaded springs, is rigid until water is admitted to the turbine and hydraulic thrust begins to act downward. Monitoring instrumentation would have to be suitable for both rigid and displaced conditions. The instrumentation would have to be capable of accurately

indicating load before, during, and after the transition from rigid to spring action.

- When the unit is operating, the springs compress and each pad tilts and displaces vertically. Instrumentation should be capable of accounting for tilt due to oil wedge formation and displacement proportional to load.

- Unlike the other types of bearings, it is possible for flexible-support bearings to experience partial degradation of its capabilities. The other types of bearings would likely experience catastrophic change in performance if a single bearing part failed. In the spring-type bearing, it is possible for individual springs to change due to fatigue or breakage. In the event of a single spring, or even a small number of springs, failing, the performance of the whole bearing would change only slightly. This is a benefit from the point of view of bearing operation, but it is very hard to detect. It would be desirable to detect a trend of small changes over time in order to avoid bearing deterioration to the catastrophic failure point.

Instrumenting the Appropriate Parameters

In order to define the function of the monitoring system, we first needed to clearly describe each of the operating modes of the turbine/generator and the condition of the bearing in each mode. In addition to the operating modes, a list of possible deviations in design, manufacture, operation, or maintenance was developed in order to further refine the monitoring system capabilities. After analysis, we decided that we want to be able to measure both the dead weight and the hydraulic thrust on the bearing. We want to monitor pad displacement and reference it to measured load. We want to compare both load and pad displacement to the generator maker's predicted results and resolve any discrepancies. We want to use startup historical data at various operating conditions as the baseline for future trouble shooting.

We decided to focus much of the monitoring system design on the bearing's transition from rigid to spring action. We feel that, if the bearing pads remain rigid too long after the unit begins rotating, the hydrodynamic oil wedge may not form, oil could be starved to the bearing surfaces, and bearing failure could result. Since we plan to shut down the oil lift system at some speed below rated speed, we want to ensure that the bearing is no longer rigid when the oil lift system is shut down. We want to verify that the pads are deflecting under minimum hydraulic thrust.

Because the pads are rigid in some modes of operation and tilting on a bed of springs in others, monitoring the bearing using strain gage or load cell data alone would be incomplete. For example, the total thrust on a bearing might be well within the design, but if the pre-load on the springs causes the pads to remain rigid after water has been admitted to the turbine, the bearing may run dry, and fail. Monitoring pad displacement alone would not be sufficient, either: pad displacement indicates only changes which occur after hydraulic thrust begins to compress the springs. By itself, it does not indicate if displacement is beginning at the correct time. Also, pad displacement measurements would likely be referenced to an arbitrary datum. Without reference to unit load, if displacement sensors were removed for maintenance and returned, there would be no way to be certain that displacement measurements were caused by the same thrust as before.

In the end we decided to monitor the thrust bearing performance by using both a load cell and a proximity probe under each bearing pad. Figure 3, items (g) and (h), show the planned location of the load cells and proximity probes. Load cells will indicate thrust values on the bearing from dead weight to the first bearing failure parameter, which is nearly twice maximum hydraulic thrust. Proximity probe gaps will be set under the pads so that the probe tips are well below the distance that the pads could displace. The zero setting of the probe will be set at the pre-load (dead weight) position of the pads. The probes will then be able to measure pad movement up when dead weight is removed and pad movement down under live load. It will measure the full range of displacement from springs unclamped to the first bearing design limit.

Expected Benefits

The benefits that New England Power expects to gain from installing a monitoring system on the thrust bearings of its newest turbine/generators are several:

- The project team expects to utilize the permanent thrust bearing monitoring system to assure correct bearing design and accurate manufacture. We will use the integration of the monitoring system into the bearing to interact with the generator maker concerning the details of the bearing design. We will use the information discovered to inspect the bearing manufacture and evaluate the maker's production quality.

- The project team expects to use the monitoring system to evaluate and manage initial unit startup. We have

established various points during the unit startup procedure for gathering thrust bearing information. Each set of data will become a baseline of performance information and each following step of the startup which might influence thrust will then be referenced back to previous data. We expect to measure unit dead weight, no-load thrust, partial and full load thrust, load rejection thrust, and runaway speed thrust. Of particular interest will be to evaluate if the bearing changes from rigid to spring action before speed-no-load and before the oil lift system is shut down.

- Maintenance personnel expect to use the monitoring system to identify and analyze changing operating conditions. After each annual inspection, major equipment modification, or unit overhaul, thrust data will be compared with past data to determine the effect of changes on thrust and thrust bearing performance.

- Maintenance personnel expect to use the system to troubleshoot thrust related problems and degradation of bearing performance. They will collect and analyze data whenever the performance of the machine might suggest a change in thrust or a change in the bearing. For instance, the system should be able to detect a change in spring constant of 2%, the failure of as few as seven out of 342 springs, or the failure of one spring clamp bolt.

- As operating experience is gained with the monitoring system, there is a potential to use it for automatic protection of turbine from thrust failures. The system will be capable of transmitting alarm relay outputs to either an annunciator or to trip relays.

Conclusions

Spring-type thrust bearings are capable of operating well in vertical shaft turbine/generators.

One of the most important points in spring-type bearing operation is the transition from rigid to spring action.

Spring-type bearings have been difficult to troubleshoot but can be monitored.

Opportunities exist in the specification, proposal, and design stages of acquiring a new turbine/generator with spring-type thrust bearings to require a bearing monitoring system.

A similar system might be used to monitor thrust problems in existing machines or to prepare for machine changes.

Electronic Governors for Stand Alone Induction Generators

Dan Levy

Abstract
Induction generators with prime movers such as hydraulic or wind turbines are often used for cogeneration. The same turbine generator configuration can be used for stand alone generation if a controller is introduced. Different controllers are needed for the doubly fed and squirrel cage induction generators. These configurations require no hydraulic or mechanical control on the turbine. The present paper describes electronic governors (controllers) for stand alone doubly fed induction generator and for the squirrel cage induction generator systems. Advantages and disadvantages are summarized.

1. Introduction

From the theory of the induction generator[1], it can be seen, that the only requirement for obtaining an output is a source of magnetizing vars and a suitable load below the power limit. The relationship between power and vars for a typical induction generator can be calculated[2]. For a squirrel cage induction generator, the required var can be supplied by a synchronous machine or by shunt capacitors. Thus an output can be obtained from this type of generator when it is connected to a system consisting only of shunt capacitors and load.

Dr. D. Levy is a Senior Lecturer with the Department of Electronic & Computer Engineering,
University of Limerick, Limerick, Ireland.

The frequency of operation of an induction generator with self-excitation is very close to the frequency corresponding to the generator speed multiplied by the number of poles pairs since the pull out slip is normally less than 10 per cent and the machine will not operate at a load above the power limit.

For constant output frequency and constant output voltage, some form of regulation needs to be introduced. This regulation can be made with the aid of controllers described in the present paper. For the doubly fed induction generator, the controller is a frequency power converter which accepts information on the speed and the output voltage from the generator stator and provides variable frequency voltage at appropriate amplitude to the rotor of the generator. In this way a feedback controller loop is established.

On the other hand, the squirrel cage generator offers no input of any kind of control; this results in increased stresses for the other parts of the mechanical and electrical system and any excess of power generated should be dissipated on an auxiliary resistive load. Furthermore, the reactive power also needs to be controlled in order to keep the output voltage and frequency constant.[3].

2.Stand Alone Doubly Fed Induction Generator

The stand alone doubly fed induction generator, together with a frequency converter, in a closed loop is able to control the output voltage and frequency independent on speed. For constant load/constant output voltage, the system provides constant torque to the prime mover independent of speed. Fig 1 shows the system arrangement. Shunt capacitors are not necessary here. Reactive vars are supplied by a battery which is charged all the time from the output of the generator through a rectifier (for the case of $\Omega < w$) or from the frequency converter (for the case $\Omega > w$). This battery drives the frequency converter which supplies variable output frequency and voltage to the rotor according to the instructions received from the controller. The controller receives a sample of the load voltage, compares it with a reference, and any difference which is an error voltage is used to modify the output voltage from the converter to the rotor in a closed feedback loop.

The controller also receives information on the speed Ω and position θ of the rotor in order to calculate the slip frequency $w - \Omega$ supplied to the rotor. Any variation in speed, will obviously result in variation

of this slip frequency in order to keep the output frequency w constant. For a fixed output voltage to the load, the output current from the frequency convector to the load is constant and depends only on the total load Z seen by the stator i.e. in complexor form:

$$i_{rotor} = F(Z) \cdot e^{j(w - \Omega)t} \qquad (2.1)$$

The rotor voltage is a linear function of the slip frequency (w-Ω) i.e.

$$e_{rotor} = [E(Z) + D(Z)\ (w - \Omega)]\ e^{j(w-\Omega)t} \qquad (2.2)$$

Therefore, the rotor power will also be a linear function of the slip frequency in the form:

$$P_{rotor} = A(Z) + (w - \Omega)\ B(Z) \qquad (2.3)$$

Rotor flux increases with the load, however, it is independent of speed. Maximum load therefore should be specified in order not to saturate the machine. The mechanical (prime mover) power is directly proportional to the speed Ω

$$P_{mechanical} = T(Z) \cdot \Omega \qquad (2.4)$$

where T(Z) is the torque. The torque on the prime mover is therefore a function of the total load Z of the stator only. If the output power from the stator is denoted by P_{out} then:

$$P_{mechanical} = P_{stator} - P_{rotor} + P_{losses} \qquad (2.5)$$

From (2.3) P_{rotor} is negative at subsynchronous speed $\Omega < w + A(Z)/B(Z)$ i.e. the battery or the stator should supply power to the rotor. Above the synchronous speed i.e. at supersynchronous speed $\Omega > w + A(Z)/B(Z)$, both the rotor and the stator will supply power. Therefore in the latter case, the frequency converter (forced commutated) and the bridge rectifier (line commutated) should be four quadrant converters (accept bidirectional power flow).

It should be noted, the total load on the stator Z is speed dependant for the system shown in Fig. 1 since the rotor load on the Converter is not constant but inversely proportional to the speed. Therefore, the equations (2.1) : (2.5) are speed dependant.
If this arrangement is integrated with a solar cell or

wind turbine in parallel with the battery, then the whole system will be an ideal, very efficient stand alone generating system.

2.1 System Analysis

The system of Fig.1 was analyzed. The load was assumed to be three phase symmetrical network. Saturation, end winding and slot effects are neglected. Iron losses are assumed constant for constant output voltage. At a steady state, the system can be described by the following equations:

For the stator $0 = (R_s + Z + jwL_s)i_s + jwMi_r$ (2.1.1)
where Z is the total load on the stator of the generator.
The output voltage $E = - i_s \cdot Z$ (2.1.2)

For the rotor :
$e_r = R_r\, i_r + (w - \Omega) \times (L_r\, i_r + Mi_s)$ (2.1.3)
The electromechanical equation:
$J\dot{\Omega} = i_r \times \phi_r - T = 0$ (2.1.4)

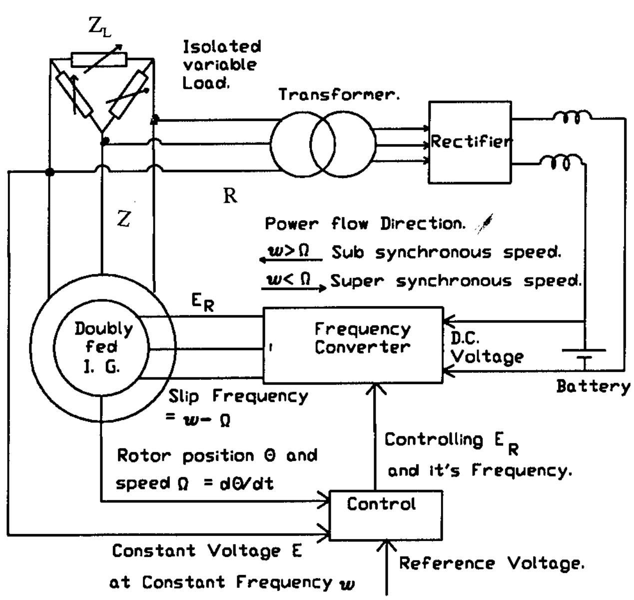


Fig.1 Doubly Fed Stand Alone Induction Generator Configuration

where ϕ_r is the rotor flux and T is the driving torque

$$\phi_r = L_r\, i_r + M\, i_s \qquad (2.1.5)$$

For a given output voltage E, the stator current is a function of Z which is in turn a function of the speed Ω. Z consists of the output load Z_L and the Converter load R (a resistive load). R is inversely proportional to the speed at subsynchronous speed, but negative at supersynchronous speed. For the system of Fig. 1 the following two sets of equations need to be added in order to close the rotor - stator loop.

$$Z = R.\, Z_L \;/\; (R + Z_L) = -\frac{E}{i_s} \qquad (2.1.6)$$

and

$$\text{Real } e_r . i_r^{*} = \frac{E^2}{R.\eta} = P_{rotor} \qquad (2.1.7)$$

where η = efficiency of the converters, E is the stator voltage, i_r^{*} is the complex conjugation of i_r. The set of equations (2.1.1), (2.1.3), (2.1.6) and (2.1.7) represents a non linear system with i_r, i_s, e_r and R as unknowns.

2.2 Numerical Calculation

Numerical Calculations per phase are shown in Figs 2 : 7 for a typical 25 kw generator for three types of load; pure resistance, resistance in parallel with a capacitor and a resistor in parallel with an inductance. The efficiency of the converter was assumed very high (close to 100%). The parameters are calculated against normalised speed Ω/w (Ω/w = 1 at synchronous speed).

2.2.1 Low Speed Operation (Subsynchronous Speed)

The speed range of operation is shown for each load. The minimum speed of operation is $\frac{\Omega}{w} = 0.8$ for a pure resistive load. Below that speed no solution for the set of equations (2.1.1), (2.1.3), (2.1.6), (2.1.7) exists. At this minimum speed, the rotor power is maximum, about 60% of the stator power Fig. 2 and therefore the efficiency is minimum, Fig. 3. At higher speed, the rotor power is reduced and therefore the generator efficiency is improved.

Rotor VAR is shown in Fig.4. It is capacitive below the synchronous speed but inductive above it. Torque/speed characteristic is shown in Fig. 5. The torque is very sensitive at lower speed due to rotor (converter) loading on the stator. This load is not

constant but high at low speed. Rotor flux is shown in Fig. 6. It increases sharply at lower speed.

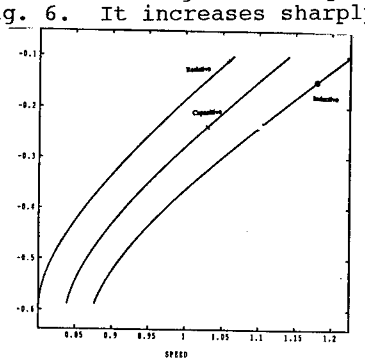


Fig.2 Rotor Power/Stator Power Ratio - Subsynchronous

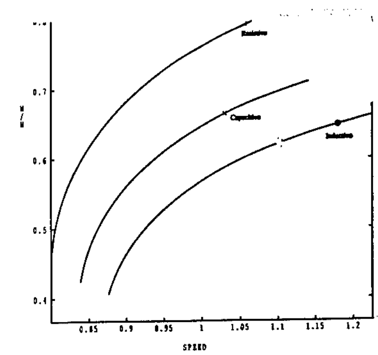


Fig.3 Efficiency - Subsynchronous Speed

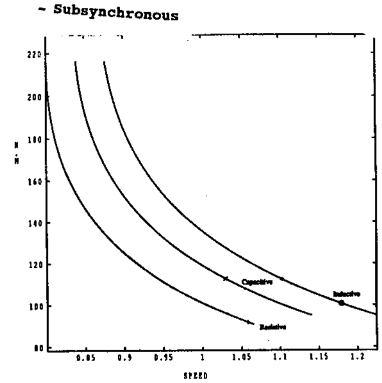


Fig.5 Torque - Subsynchronous Speed

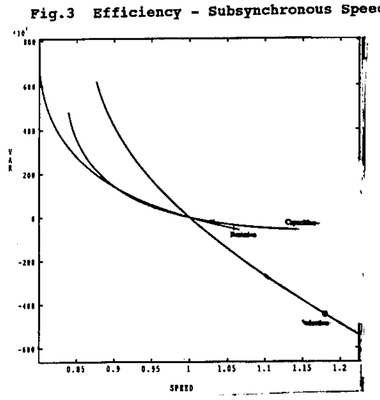


Fig.4 Rotor Var - Subsynchronous Speed

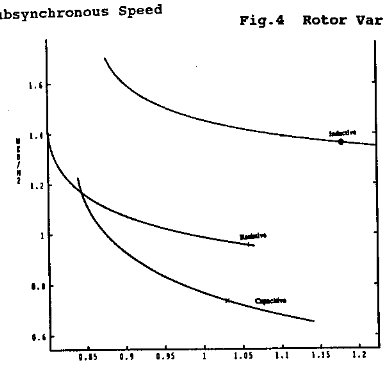


Fig.6 Rotor Flux - Subsynchronous Speed

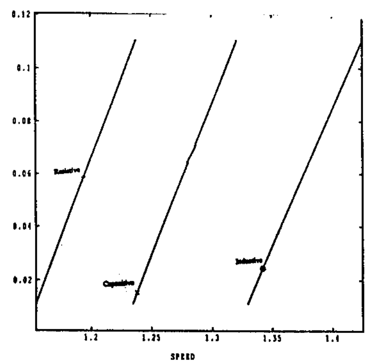


Fig.7 Rotor Power/Stator Power Ratio Supersynchronous Speed

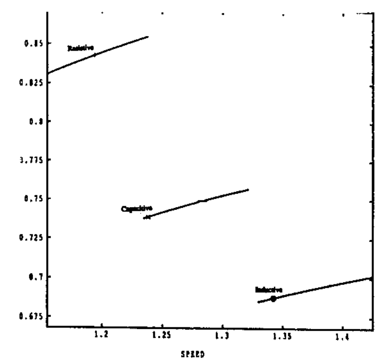


Fig.8 Efficiency - Supersynchronous Speed

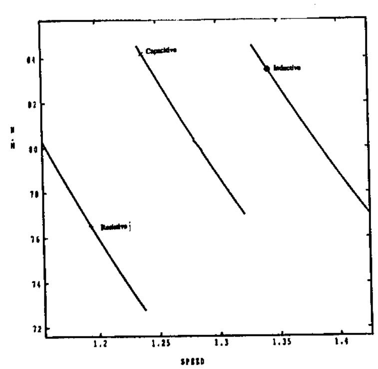


Fig.10 Torque - Supersynchronous Speed

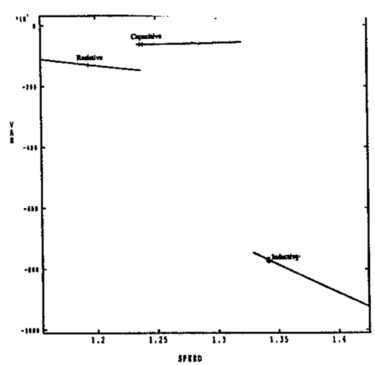


Fig.9 Rotor Var - Supersynchronous Speed

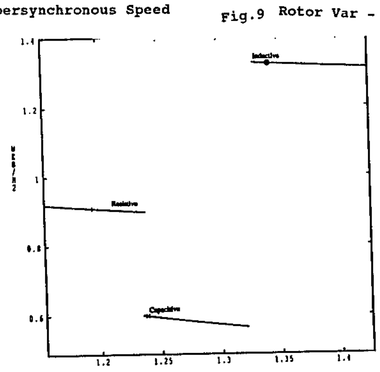


Fig.11 Rotor Flux - Supersynchronous Speed

2.2.2 High Speed Operation (Supersynchronous Speed)

The minimum speed for this mode is shown in Fig. 7 where both rotor and stator are supplying power to the load. The rotor share of power increases with speed. The efficiency Fig. 8 improves with speed. It is higher at higher speed operation. Rotor Vars Fig. 9 are inductive but minimum for a capacitive load (i.e. for the amount of capacitance used in this example). Fig. 10 shows the torque/speed characteristic which is very linear in contrast to low speed operation. Rotor flux Fig. 11 is almost constant but decreases slightly at higher speed.

3. Stand Alone Squirrel Cage Induction Generator

The impedance controller shown in Fig. 8 provides reactive power for the excitation of the generator and the system load. Adding a resistor (heating elements) and a thyristor bridge enables one to control the real power. This results in the control of the turbine speed without the use of hydraulic turbine controls [4].

The impedance controller takes the difference between the power provided by the uncontrolled turbine and the power absorbed by the generator load. Consequently constant speed is maintained.

The three phase thyristor bridge shown in Fig. 8 is basically a three phase naturally commutated full bridge A.C. to D.C. converter. The chopper can be regarded as a regulator (D.C. to D.C. regulator) with chopping frequency relatively much higher than the converter (thyristor bridge).

The real and reactive power are controlled separately by adjustment of the thyristor delay angle α and the chopper pulse width PW[5]. These can be calculated from solving two sets of equations with the aid of a microprocessor in real time:

$$\text{Real Power} = f(\alpha, PW) \quad \ldots(3.1)$$

$$\text{Reactive Power} = g(\alpha, PW) \quad \ldots(3.2)$$

therefore, the chopper is necessary to decouple the control between real and reactive power. The chopper is synchronized to the 60 degrees conduction periods of the bridge and is operated with multiple chopping periods during each 60 degrees conduction period. Synchronization and multiple chopping, reduces voltage distortion (better filtering).

Equations (3.1) and (3.2) also means the output voltage and output frequency of the induction

generator can be decoupled and controlled separately i.e.

$$\text{Voltage} = V(\alpha, PW) \tag{3.3}$$

$$\text{Frequency} = F(\alpha, PW) \tag{3.4}$$

α and PW will therefore control the thyristor bridge and the chopper respectively. The torque/speed characteristic of the generator will therefore be constant for a given constant load. The prime mover power will be proportional to the speed and so will the power dissipated on the heating element if the load is left constant.

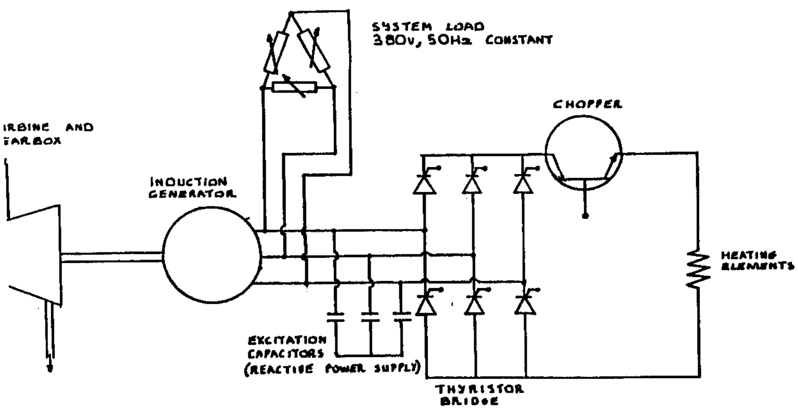


Fig.12 Squirrel Cage Stand Alone Induction Generator Configuration

4. Conclusion

A doubly fed induction generator when integrated with a forced commutated frequency converter, a line commutated converter and a battery, can be used as a stand alone generator. This configuration is able to provide constant output voltage and frequency, at higher efficiency, to an isolated load. Therefore, no hydraulic or mechanical control on the turbine is needed. When the generator is operated at supersynchronous speed, higher efficiency with linear torque speed characteristics are obtained.

Any configuration with the squirrel cage machine as stand alone generator with constant output voltage and frequency independent of the load and turbine speed, will result in a low efficiency system since any excess of power needs to be dissipated on an auxiliary resistive load. If such dissipated power has no use, for example, heating, this type of system is not practical for general application.

REFERENCES

1. Barkie and Ferguson, Induction Generator Theory and Application, AIEE Transactions Vol. 73, Feb. 1954, pp. 12-19.

2. Gary L. Johnson, Wind Energy Systems, Prentice Hall, N.J. 1985, pp. 252-263.

3. Progress report of the Technical University "Braunschweig" in Holland, Proceedings Wind Energy R and D contractor meeting Alghero, 3 - 5 June 1992, Joule 1990 - 1992, R & D Programme in the field of non-nuclear energy and rational use of energy pp. 151-152.

4. R. Bonert and G. Hoops, Stand alone induction generator with terminal impedance controller and no turbine controls, IEEE Transactions on Energy Conversion, Vol. 3, No. 1, March 1990, pp. 28-31.

5. D. Levy & E. McQuade, Analysis & Synthesis of Static Power Converters, IEE Proceedings - G on Electronic Circuits and Systems, Vol. 133 No. 1, pp. 39-57, Feb. 1986.

Hydro Generator Winding Specification Trends

D.G. McLaren [1] W. Newman [2] R. H. Rehder [3]

GE Canada General Electric GE Canada

Abstract

This paper discusses specifications written for windings of hydraulically driven generator. Currently popular tests such as voltage endurance, thermal cycling, partial discharge and surge testing are described and commented upon.

Introduction

Generator specifications are being written with greater definition of test requirements to enhance the future reliability and life of the stator windings. Specifications are now including voltage endurance life, thermal cycling effects, partial discharge testing, and surge voltage requirements. In some cases the limits used may have a major impact on the size and cost of a machine without extending the useful life of that machine.

Voltage Endurance Specifications

Voltage endurance tests are common practice in North American specifications but are not so often used elsewhere unless the specification had its roots in North American practice.

1. Supervisor, Electrical Engineering, Hydro Operations, Power Systems and Services, GE Canada, Bldg. 30/2, 107 Park St. N., Peterborough, Ont., Canada, K9J 7B5

2. Senior Design Engineer, Apparatus Service Department, General Electric Co., Bldg. 6–2nd Floor, Schenectady, N.Y., U.S.A., 12345

3. Technical Counsellor, Insulation Systems, Engineering Laboratory, Power Systems and Services, GE Canada, Bldg. 100, 107 Park St. N., Peterborough, Ont., Canada, K9J 7B5

As used in North American specifications, the purpose of the requirement is to ensure a minimum quality requirement on the procurement. A small sample must pass a certain number of hours of testing without failure.

In its other usages, voltage endurance testing is used as a valuable tool for comparing insulations.

There is not yet a clear consensus as to suitable test requirements. While 250 hours at 35 kV and 400 hours at 30 kV are often applied to 13.8 kV windings, the temperatures, quantities and sample sizes may vary according to the customer or manufacturers practices. Each of these have different effects on the test's expected results.

In the IEEE test [1], the test has a single accelerated factor. Temperature is not generally too extreme with usual values being in the range of 90 to 120 degrees. C. In that range, there is not too much shift in expected times to failure.

Voltage has a direct effect on test results since it increases or decreases the applied stress. The variation of expected results with stress may fit either a power or an exponential model as described in IEEE 930 [2].

A second, and less considered effect is that of sample size. There are two variations inside this issue.

Firstly, one can test more coils. If there are more coils tested, then there is a greater probability of an early failure occurring. Secondly, the individual sample coil may be larger or smaller than previously tested. A coil a mile long would fail in a very short time where a short piece would last a long time. The stress distribution in a coil is highest at the edges of the coil and so the failure times are shorter there. One result of this is that the time to failure depends more on the length of the edge than the area of the insulation.

The manufacturers approach is to look at their system's capability of time to failure versus stress and arrange that there is a certain probability of failing the test. Each one usually has a significant amount of data describing how their life versus stress varies at or near such test levels. Additionally they should routinely test their product to make sure it is at least as good as past experience and for process control evaluation.

Assuming an inverse power model, the stress–life relationship may look like the following:

$$\alpha \;=\; \textit{Characteristic Life} \;=\; A \, x \left(\frac{kV}{mm} \right)^{-n} \qquad (1)$$

Let A = 2,633,000 hours and n = 3.049 for reasons given below.

Since the manufacturer will have tested the system previously, they will have a reasonable idea as to the value of β for that system. In this example we will let $\beta = 1.5$. Assume also that this data is based on test specimens 14 inches long (L_o = 14 inches).

The manufacturer can now express α as a function of build once the test voltage is specified.

After that, the probability of a sample of length L (inches) passing N (hours) of testing is

$$R(t) = e^{\frac{-L}{Lo}\left(\frac{t}{a}\right)^{\beta}} \qquad (2)$$

The parameters in this example have been rigged to set the failure probabilities of the 30 and 35 kV test (400 and 250 hours respectively) at 0.95 for a single 15" specimen.

Consider the following specification test requirement applied to this insulation system: 35 kV, 250 hours, 4 full legs, 80" test length, no failures before 250 hours. For two builds, the following results are obtained:

Build (mm)	Operating Stress (kV/mm)	Test Stress (kV/mm)	α Test (hours)	R(t hours) (250 hours)	R^4
3.0	2.66	11.67	1470	0.67	0.20
3.4	2.34	10.29	2153	0.80	0.40

Now consider another variation where the number of samples is increased to eight. While this brings statistical improvement, the odds of contractual failure increase exponentially.

Build (mm)	Operating Stress (kV/mm)	Test Stress (kV/mm)	α Test (hours)	R(t hours) (250 hours)	R^8
3.0	2.66	11.67	1470	0.67	0.04
3.4	2.34	10.29	2153	0.80	0.16
3.8	2.10	9.21	3023	0.87	0.33
4.2	1.87	8.33	4101	0.92	0.50

A significant improvement in expected test probability occurs as the applied stress is adjusted by adding insulation.

The values of A, n and B used in equations 1 and 2 above will depend on the insulation system and manufacturing processes used and so will be different for each product.

An alternate system with a flatter slope will have much better probabilities below the line. It will also show a sudden threshold, above which the probability of passing a test will be nearly zero.

Adding insulation means subtracting copper and adding temperature for most windings so the point of this demonstration is to show that while the test criteria does indeed cause the vendor to be conservative, it can also cost the user losses and some temperature . The calculation of how much loss or temperature it costs depends on the size and winding design of the specific unit.

Certainly making insulation thicker compensates for some problems, but applications and systems should also be evaluated using their service experience.

Thermal Cycling

Some specifications are now calling for thermal cycling tests. The reason for these tests is to verify that the insulation system will not have premature failures due to delamination or other effects resulting from thermal cycling operations such as pumped storage applications. At this time some specifications are using a paper published in IEEE in 1992 [3] as a description of the test procedures. The IEEE has a Working Group preparing a Recommended Practice for Thermal Cycle Testing (P1310) and the IEC SC2J/WG 6 is developing a document on thermal cycling. This testing procedure is in its infancy and until a sufficient number of tests have been performed, it will not be known just how closely the test results match actual field performance.

Specifications have varied but the tests involve heating the bar or coil to a specific temperature, say 150 degrees C, in 30 to 60 minutes and then cooling to 40 degrees C in 30 to 60 minutes, and repeating this procedure for 250 to 500 cycles. At specific intervals, measurements of Dissipation Factor, Tip Up, Capacitance and physical dimensions are made. The general assumption at this time is that the measurement results are to stabilize as the test progresses if the insulation system is to be acceptable. A typical curve of Dissipation Factor versus Number of Cycles is given in Figure 1. Some specifications call for voltage endurance tests after thermal cycling with specific minimum hours to failure at a specific voltage. The industry does not have enough experience to understand completely the effects of thermal cycling on the voltage endurance test life. A manufacturer may increase insulation thickness to reduce the risk of failing a very long voltage endurance time when in actual field applications there is more than sufficient insulation on the original insulation build. The increased insulation increases machine size and cost.

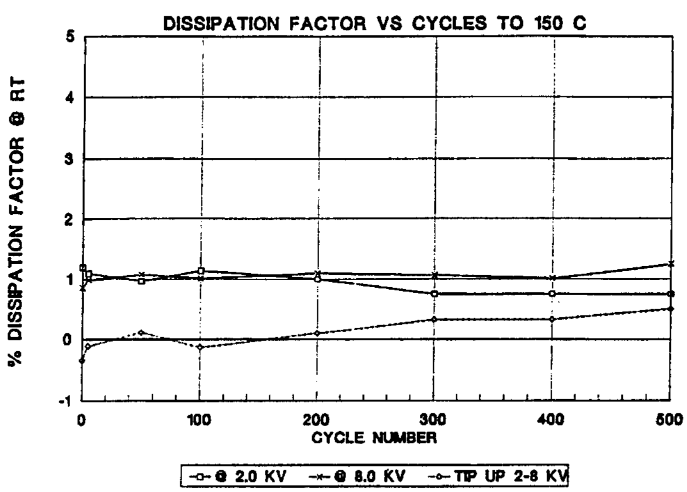


FIGURE #1

These tests require space, a high current power supply to heat the bars, and blowers to force cool the assembly. The time involved is approximately 2 months for a 500 cycle test, including the measurement tests that must be done after intervals such as the 50th, 100th, 200th, 300th, 400th and 500th cycle.

Therefore the tests involve considerable expense. Specifying that these tests be done in advance of production, or from samples taken from production, can have an impact on manufacturing and shipping schedules and should be addressed when writing the specifications.

Some standards are advocating tests using bars or coils mounted in full stator slot sections. This can make testing very costly and time consuming as compared to the present practice of performing the tests with the coils or bars unrestrained and in free air. Some companies are advocating simultaneous thermal cycling and voltage endurance but there can be a complication in interpreting results with this arrangement as the failure mechanisms of the two procedures compete and combine.

With time, industry standards will be issued and this should stabilize the test procedures. Also, as more testing is done, there will be a closer tie between test results and insulation system field performance and specification limits for these tests can be more realistic.

Partial Discharge Analysis

Partial Discharge Analyzers (PDA) have been applied to Hydro Generators at site for a number of years. Operational people have become skilled in the interpretation of the information provided by these analyzers and many Utilities establish maintenance and repair programs based on this information. Such diagnoses should be verified by physical evidence or alternate tests as may be required.

There is now a trend to ask for partial discharge measurements on the bars or coils along with the other tests such as voltage endurance and thermal cycling. The difficulty is again interpretation of results. First of all, there is no industry agreement on how the measurements should be made. The variables are the frequency response of instrumentation (pulse width), pulse or apparent corona magnitude, pulse counts, and the relationship of the pulse to the power frequency in time. Sorting out noise from real partial discharge within the bar or coil can result in a loss of critical discharge information. Partial discharge activity is time related and is sensitive to preconditioning. For example, tests made on a new bar when repeated a day later may not show the same results. If tests are done after the bar has been preconditioned by a high potential test, the partial discharges will probably have decreased in number. The reason for this could be due to the voids having been carbon coated or gas pressure being developed in the void as a result of the initial discharges caused by the high voltage. This is similar to the mechanism than gives us lower dissipation values after high potential tests.

What voltage should be used for partial discharge tests? The mechanism of failure of the insulations under partial discharge activity is a function of the applied voltage and the type of discharge that is occurring. Lower voltage glow discharges cause different chemical material changes than do avalanche high energy spark discharges. When the purpose of the test is to ensure some measure of long life and reliability, we must relate test results to a mechanism of failure that corresponds to the mechanism during normal machine operation.

Measurements at site can use capacitive couplers or special detectors located under the wedges in the slots containing line end coils.

Electromagnetic interference measurements using radio frequency current transformers in the neutral connections of the operating machine at site is another method of partial discharge detection that is being developed. Patterns in the frequency spectrum contain information relevant to discharge location.

There are many Laboratories around the world now trying to resolve some of the questions raised relative to PDA testing and interpretation of results. Any specifications requesting PDA tests will have to be reviewed carefully with the manufacturer to ensure there is a mutual understanding of how measurements are to be made and the interpretation of results. All insulation systems do not have the same PD response under identical test protocols.

Surge Testing

Surge Testing is generally applied to multi turn coils. Voltage surges whether created by a lightning strike on connecting power lines, by switching surges, or by electronic switching, will apply voltage stress between turns, particularly on line end coils. The new IEEE Standard 522 [4] describes the test procedures and recommended levels. The IEC Standards [5] have comparable test values. These can be referred to in specifications. The tests are non–destructive and are usually done on individual coils prior to winding connections being added. Testing of a complete machine winding may be a problem from an equipment availability standpoint as charging capacitance may be high.

Bar type windings and the first line end coils of multi turn coils will experience the same surge voltage stress to ground as turn to turn but usually the ground wall insulation is so very much thicker than the turn to turn insulation that surge testing is not critical. It is not usually specified for bar windings.

Surge testing usually applies to fast front (0.1 to 0.2 microsecond rise time) surge voltages caused by switching surges. Surges due to lightning

strikes normally have a slower front (1.2 microsecond rise time) and the test level and withstand capability of the insulation is higher. Tests can be done with the slower front but modern insulation systems usually have good margins and these tests are not normally considered.

Surges due to electronic switching associated with variable speed drives are considered to be fast front (0.1 to 0.2 microsecond rise time) but of a lower magnitude than switching surges. A 1.8 per unit value is generally accepted. The surges occur more frequently, 12 surges per cycle or 720 per second and therefore there is a question of surge voltage endurance. There is work being done on surge voltage endurance but there are no standards or references at this time that could be used in specifications.

Summary

The latest trends in specification test requirements are leading the way to significant changes in bar and coil design which have not been totally proven to substantially increase machine life. Over the next several years, as the specifications, standards and test methods stabilize, there will be better information to evaluate the actual improvements gained in machine life extension.

Bibliography

[1] IEEE 1043–1989 IEEE Recommended Practice for Voltage–Endurance Testing of Form–Wound Bars and Coils

[2] IEEE 930–1987 IEEE Guide for the Statistical Analysis of Electrical Insulation Voltage Endurance Data

[3] Stone, G.C., Lyles, J.F., Braun, J.M., Kaul, C.L., A Thermal Cycling Type Test for Generator Stator Winding Insulation, IEEE Transactions on Energy Conversion, Vol. 6, No. 4, December, 1991.

[4] IEEE 522–1992 IEEE Guide for Testing Turn–to–Turn Insulation on Form–Wound Stator Coils for Alternating Current Rotating Electrical Machines

[5] IEC 34–15–1990, Part 15, Impulse Voltage Withstand Levels of Rotating A.C. Machines with Form–Wound Stator Coils

Air Blast Circuit Breaker Replacement Arc Chutes

Charles Rinck[1]

Abstract

This paper describes the development of prototype, composite material arc chutes to replace deteriorating original equipment arc chutes in existing General Electric type AR-A1, 15 kV, 2500 MVA air blast circuit breakers. Design and tests were developed by The Dalles - John Day Project and the Hydroelectric Design Center of the U.S. Army Corps of Engineers, North Pacific Division.

Introduction

The powerhouse at John Day Dam, on the Columbia River, first began generating power in 1968. There are 16 main unit generators, each with an installed capacity of 155 MW. The main unit circuit breakers are General Electric, type AR-A1-14.4-2500 air blast circuit breakers which use forced air cooling to deliver 7200 Amp.

After several years of operation, the insulating material used in the circuit breaker arc chutes started to split and warp. Fragments of this material would fall off, sometimes in the area around the stationary main contacts. The guide blocks also became misaligned to the point where the moving contacts were not always directed straight into the stationary main contacts.

----

[1]Chief of Engineering & Contracts Section, U.S. Army Corps of Engineers, The Dalles - John Day Project, P.O. Box 564, The Dalles, OR 97058

This paper was presented at the Doble Engineering Conference in Boston, MA; Permission for its use was given by Doble Engineering Company.

Original Equipment Material Problems

The original equipment arc chutes were fabricated from a laminated, vulcanized cellulose. This material has good dielectric strength and is machinable. Moreover, it ablates cleanly in the presence of an arc, contributing in the arc interruption process along with energy dissipation through parallel resistors and arc elongation from a high pressure air blast. However, in the warm and dry environment of a 7200 full load amp circuit breaker, the vulcanized cellulose loses moisture and parts begin to shrink; some more than others and not necessarily uniformly throughout the laminations. Adjoining parts no longer fit snugly together and are allowed to migrate to where they may eclipse the path of the moving contact blades, either obstructing or diverting blade motion or having fragments sheared off during the closing operation. "Slab" parts, such as arc chute side plates, may cup and bow, resulting in non-uniform clearances, binding and misalignment.

Initial Solution

The initial response from our Maintenance Section was to repair delaminated parts using adhesive and to correct misalignment during semi-annual maintenance outages. In cases of extreme deformation or delamination, complete replacement arc chute assemblies were procured. But, parts which had been repaired with adhesive inevitably failed again in service and replacement arc chutes had long lead times.

Alternative Solution -- Redesign

Our goal was to develop a set of drawings so that arc chute parts could be fabricated from an electrically equivalent but mechanically superior material, in our machine shop or by a private contractor.

The 30 or so parts of an original equipment arc chute were measured with a vernier caliper and preliminary sketches were drawn. Slight adjustments were made to the dimensions and tolerances, with the goals of achieving a snugger fit and ease of assembly.

The first choice of a replacement material was "Glastic", grade UTR, a NEMA Grade GPO-3 fiber glass reinforced polyester laminate. "Glastic" was chosen primarily because of its high dielectric strength, long carbon tracking resistance and low water absorption.

Fabrication Problems

One problem with "Glastic" was that it was not available in thicknesses greater than 1-1/4 inch. Consequently, thicker arc chute parts had to be machined from two slabs joined together with adhesive. Epoxylite #223 epoxy adhesive was recommended by the local manufacturer's representative.

Another problem was that "Glastic" was not as easy to machine as vulcanized cellulose. Edges often had a rough texture and could irritate the skin of people handling the arc chute parts. It was also difficult, if not impossible, to achieve some of the dimensional tolerances called out (as small as $\pm$ 0.003 inch) without using a numerical controlled milling machine.

Early Prototype Testing

Enough parts for three complete arc chutes were fabricated in-house. One of the arc chutes was assembled with temperature labels attached to some of the internal parts. The arc chute was then placed in one phase of an operating breaker and exposed to progressive current interruptions at 3400, 4400, 6400 and 7200 Amp. The arc chute was inspected internally between interruptions and then removed and disassembled after the final interruption. There was only a light discoloration of the "Glastic" where it had been exposed to the arc. No evidence of tracking or ablation was apparent. The highest detected temperatures were on the order of 160° C (320° F) for parts immediately adjacent to the arc.

Modifications

The epoxy adhesive did not perform satisfactorily. With the shock loading present during breaker operation, some of the laminated parts developed small cracks in the bonding layer or split altogether. We changed to 3M modified epoxy structural adhesive #2216 B/A, which has since proven satisfactory in service.

We also changed a dimension on one of the support blocks adjacent to the auxiliary contact assembly. This was done to provide a bracing support surface for the copper channel which retains the arcing contacts of the auxiliary contact assembly.

Testing per ANSI Standards

The prototypes endured full load interruptions and several years of service without difficulty. However, no testing had been performed which would validate the new arc chutes' suitability for interruption of 100,000 Amp. at rated voltage or maximum interruption of 120,000 Amp, according to the AR-A1 nameplate figures.

Jack Nelson, an electrical engineer with the Corps' Hydroelectric Design Center in Portland, Oregon, reviewed the ANSI standards for testing of high voltage circuit breakers and wrote the specifications for testing to be performed by an independent laboratory, using one of the main unit generator circuit breakers from John Day Dam.

The testing was performed at the General Electric Company Skeats High Power Laboratory in Philadelphia. Besides a "Glastic" and a vulcanized cellulose fiber arc chute, a NEMA Grade C cotton cloth phenolic arc chute was also tested. Arc chutes of these three materials were installed in the John Day "test" circuit breaker, which was then subjected to carrying, closing and opening currents ranging from zero to full rated short circuit Amps, for minimum and maximum rated voltages.

The results showed that the phenolic arc chute began to fail by charring, when currents as low as 5000 Amp. were interrupted.

The "Glastic" arc chute started to fail at the limits of the AR-A1's nameplate ratings. The breaker failed to interrupt 120,000 Amp. after a 190,000 Amp. momentary (close and latch and mechanical withstand) test. A second "Glastic" arc chute was able to successfully interrupt 120,000 Amp. but then failed on a combined test of closing at 190,000 Amp. followed by opening at 120,000 Amp.

The original equipment vulcanized cellulose fiber arc chute interrupted all of the test currents successfully.

Detailed Findings

Although there was ablation evident in both the vulcanized cellulose and "Glastic" arc chutes, the degree of ablation and appearance of the affected surfaces was very different.

In "Test Report, Prototype Arc Chute Short Circuit Tests, July - August, 1984" (Nelson, 1985), the interior of the "Glastic" arc chute was described as being "extensively contaminated with molten contact material in splatters,

droplets and in incompletely-vaporized fine reddish dust. There was also a lot of black dust, apparently mostly re-condensed vaporized contact material. The black and red dust could also possibly contain 'Glastic' particles, including carbonized polyester and fused glass. The 'Glastic' surfaces also contained fine black particles and some small black molten streamers, which were assumed to be fused glass."

For the vulcanized cellulose arc chute, the ablation was greater but the surfaces were smooth and clean, with no evidence of and high-temperature arc products.

Composite Material Arc Chutes

With the goal of achieving the "clean ablation" characteristics of the vulcanized cellulose arc chute and the mechanical stability of the "Glastic" arc chute, Jack Nelson developed a composite design.

Internal parts exposed to the full energy of the arc were made of vulcanized cellulose. In cases where these parts were exposed to shear forces parallel to the laminations (because of stress concentrated in milled slots or threaded holes for screws), the parts were reinforced with threaded, vulcanized cellulose dowels, inserted normal to the plane of the laminations.

Parts that were not directly exposed to the arc were made out of "Glastic".

The arc chute side plates and top plate were fabricated out of "Glastic" for structural reasons but were given a cladding of vulcanized cellulose in areas exposed to the arc.

Present Situation

Since the testing at the high power laboratory in 1984, we have had about six composite material arc chutes fabricated by private contractors. All the contractors have used numerical controlled milling machines; the latest firm has used CAD/CAM software and equipment.

We have refined our fabrication drawings, at the fabrication contractors' recommendations, so that parts are more easily machined and dimensional tolerances are more reasonable. A couple areas of poor fit have become apparent in this process and dimensions have been revised, accordingly.

There has not been any indication of delamination or fracturing for the composite material arc chutes which have been in service over the past five years.

We are not planning to re-test the composite material arc chutes at a high power laboratory.

Conclusion

We have developed a composite material arc chute which we feel will safely interrupt the air circuit breaker nameplate rated currents, while providing a greater degree of mechanical stability, over time, than the original equipment material.

Appendix

1. Nelson, Jack M., "Test Report, Prototype Arc Chute Short Circuit Tests, July - August, 1984."

DOUBLE RUNNER FRANCIS TURBINES
FOR APPLICATIONS WITH WIDE FLOW VARIATIONS

D. Robert[1], J. Nesvadba[2]

Abstract

The paper describes the advantages of the use of horizontal double runner Francis turbine versus the classical solutions of single runner Francis turbine either horizontal or vertical, in the range of heads and discharges applicable to small hydro power plants.

The independently controlled guide vane apparatus (for lower heads) or ring gate (more efficient at higher heads) permits closing of one-half of the double runner turbine and is used when good efficiencies at part load operation are essential to the economical feasibility of the project.

With one unit only, this solution gives the same partial load performances as the classical solution of two units with single runner Francis turbines.

Historical Background

Soon after the invention of Francis turbine, multiple runner turbines were introduced in order to handle large flows while containing the dimensions of turbine components.

In these early stages most of the machines were operating under low heads in an open flume arrangement. The double runner machines were mostly of the camelback type whose advantage was to have only one draft tube for two runners.

----

[1]Marketing Manager, Neyrpic Minihydro, Pont de St. Uze, France
[2]Technical Director, Neyrpic, Inc., Shelton, CT

A large number of open flume camelback turbines were installed, many of them having several twin turbines on the same shaft and some are still in operation today. After its invention around World War I, Kaplan turbines replaced open flume Francis turbines for low head operation and the main hydraulic development of Francis turbines was directed to higher heads. Consequently, for the double runners turbines, the double discharge type was preferred over the camelback which required two spiral case inlets and presented poor efficiency due to the bad mixed flow pattern in the common single draft tube. In fact, very few camelback turbines with spiral cases were installed.

Meanwhile improvements in technology resulted in a growing trend towards larger and larger turbines which led to more and more development of vertical axis Francis turbines. The shaft deflection limited the size of horizontal axis Francis turbines at approximately 30 MW.

After World War II, it was commonly acknowledged that large Francis turbines should have a vertical axis and small unit should be of the single runner horizontal axis type. Some utilities, however, still favoured the double runner double discharge Francis turbines for small and medium size units due to its advantages and today there is a growing interest in this arrangement.

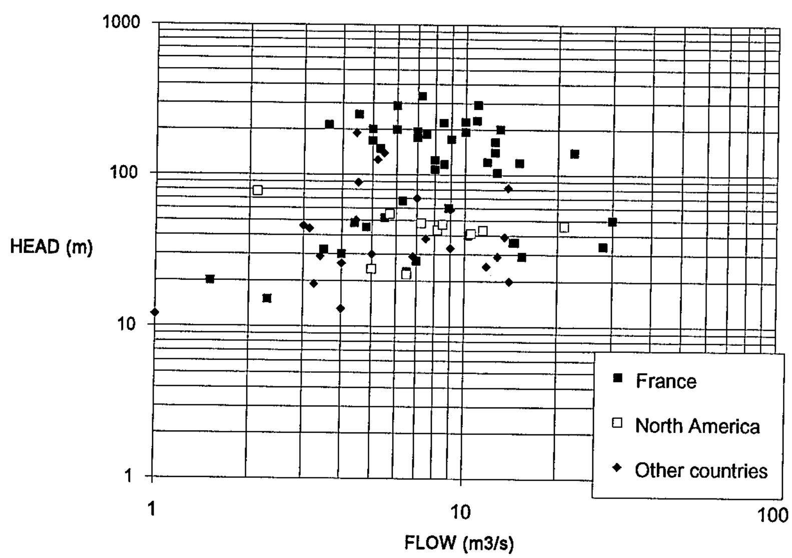


Fig. 1 - Range of Application of Double Runner Francis Turbine

Comparison Between Double and Single Runner Turbines

For the same range of available flows, the double runner turbine presents two technical advantages over the single runner turbine.

Firstly, it has less sensitivity to cavitation. Since each runner takes only half of the flow, the specific speed is divided by $\sqrt{2}$ and therefore the critical sigma is lower. This means that for the same safety margin of the cavitation free operation, one has the choice either to run at a higher speed with the same setting, or at the same speed with higher turbine setting. The former solution leads to a less expensive generator, the latter to savings in civil works costs. The choice is made by optimization of the overall project particulars.

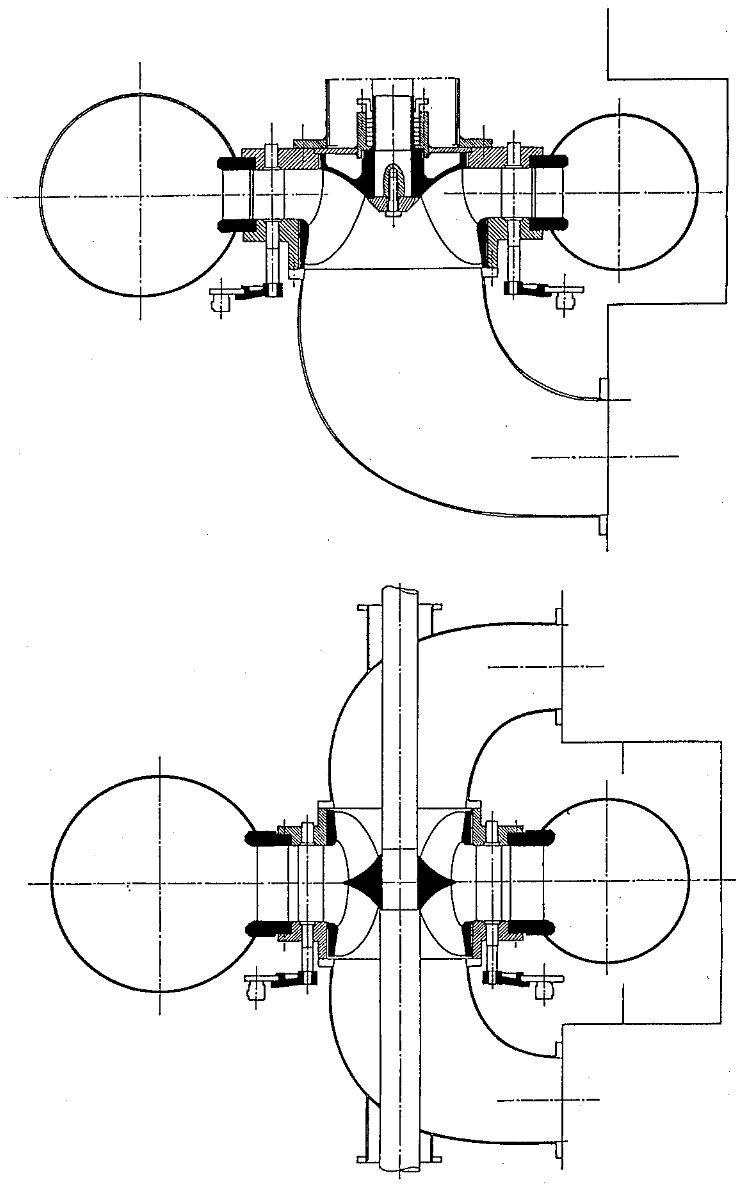
Secondly, since the discharges of each runner are in opposite directions the hydraulic thrusts are balanced and this decreases the thrust load and the cost of the bearings.

Another point worth noting is that the manufacturing cost of a double Francis runner turbine is only a few percent higher than the cost of a single runner turbine of the same capacity.

The double runner outlet diameters are approximately the outlet diameter of the single runner turbine divided by $\sqrt{2}$, therefore the distributor pitch diameter and the radii of the spiral casing centerline are reduced more or less by the same factor, while the height of the distributor is multiplied by around $2/\sqrt{2}$ and the inlet diameter of the spiral case remains the same.

The overall account of all these dimension related costs, along with change in runner costs, longer shaft and two smaller draft tubes, is nearly balanced.

Fig. 2 compares space requirements of single and double runner Francis turbines of the same capacity (4400 kW under 44 m head) using the same hydraulic design with a specific speed at best efficiency point of 214 (in metric units). The double runner has the outlet diameter of 925 mm, a speed of 600 rpm and was used for Black River project in Canada. The single runner would have had a 1275 mm runner diameter and a speed of 400 rpm.



Single Runner Francis Turbine
4400 KW, 44m Head, 400 RPM
1275mm Runner Dia.

Double Runner Francis Turbine
4400 KW, 44m Head, 600 RPM
925mm Runner Dia.

Fig. 2

Comparison Between One Double Runner Unit and Two Single Runner Units

In this case all the runners have the same specific speed and the optimization is reduced to the choice between one larger unit and two smaller units.

The relevant point is that the requirement in space for the lay-out of a double runner turbine is hardly larger than the one needed by one machine of half the capacity and consequently far less than for two units. The powerhouse dimensions are reduced, leading to important saving in civil works costs.

From what has been said of the comparison of cost for double and single runner turbines of the same capacity, a double runner turbine is definitively less expensive than two single turbines of half the total capacity each.

It is known that an appreciable savings also results from the use of only one set of larger capacity electromechanical equipment such as the inlet valve, generator and electrical controls, instead of two.

The penstock wye branch is also eliminated. The only disadvantage in the cost of the one double runner unit is that the generator is heavier and the powerhouse crane must have a larger capacity.

Some customers may object that with only one installed unit there is no power generation at all during maintenance. Today, this can be mitigated with the general practice of preventive scheduled inspections and maintenance during the low flow periods.

Particular Applications for Wide Flow Variations

The Francis turbine is not very flexible to accommodate large flow variations because of the hydraulic instabilities that develop in the draft tube around 40 % of nominal flow and poor efficiency at lower flows.

More and more in small hydropower the short term return on investment is the main aim. In many cases, this permits the installation of only one single unit because costs for civil works and electrical equipment for two units would be unaffordable. Then, depending upon the flow duration curve shape, it could happen that the partial load performances of one unit only may not allow an economical development of the project because of insufficient yearly power generation.

Such a problem of part load operation can be solved by the possibility of using only one half of the double runner Francis turbine.

There are two solutions for the part load operation. Depending upon the head, either by shutting off one of the independently controlled guide vane apparatus used for lower heads or by shutting off the ring gate-used for higher heads.

Double Runner Francis Turbine with Independent Controls

The independent controls of the guide vanes allows the closing of one half of the double runner at partial loads.

Figure 3 shows the guide vane apparatus details. This is a double runner horizontal Francis turbine in which an intermediate ring is fixed inside the stay-ring by spacer bolts. This ring is located in the plane where both halves of the double runner are joined by their crowns. It extends up to the runner crown, forming a middle wall inside the guide vane apparatus. The guide vanes are split into two halves, each having a height corresponding to the inlet opening of the runner halves. Each half guide vane has an outer trunnion linked to an external operating ring and an inner trunnion whose bearings are housed in the intermediate ring.

Starting from the fully opened position of both sets of guide vanes, the two operating rings are closing together in the same rate as the flow decreases.

When reaching slightly less than 1/2 of the rated flow the hydraulic control unit of the turbine automatically closes one set of guide vanes and opens the other one. As the flow decreases further, the control unit adjusts the second set of guide vanes to the corresponding position.

A labyrinth ring mounted on the intermediate ring facing the runner crowns, limits the leakage when operating with only one runner. Efficiency curves have similar shapes as those of two identical turbines. Values are almost the same when operating both runners. With only one runner in operation, i.e. at flows lower than 1/2 the total capacity, the efficiencies are lower than those obtained when one of the two identical units is used by around one percent due to the leakage through the runner crown labyrinth and thru the closed set of guide vanes.

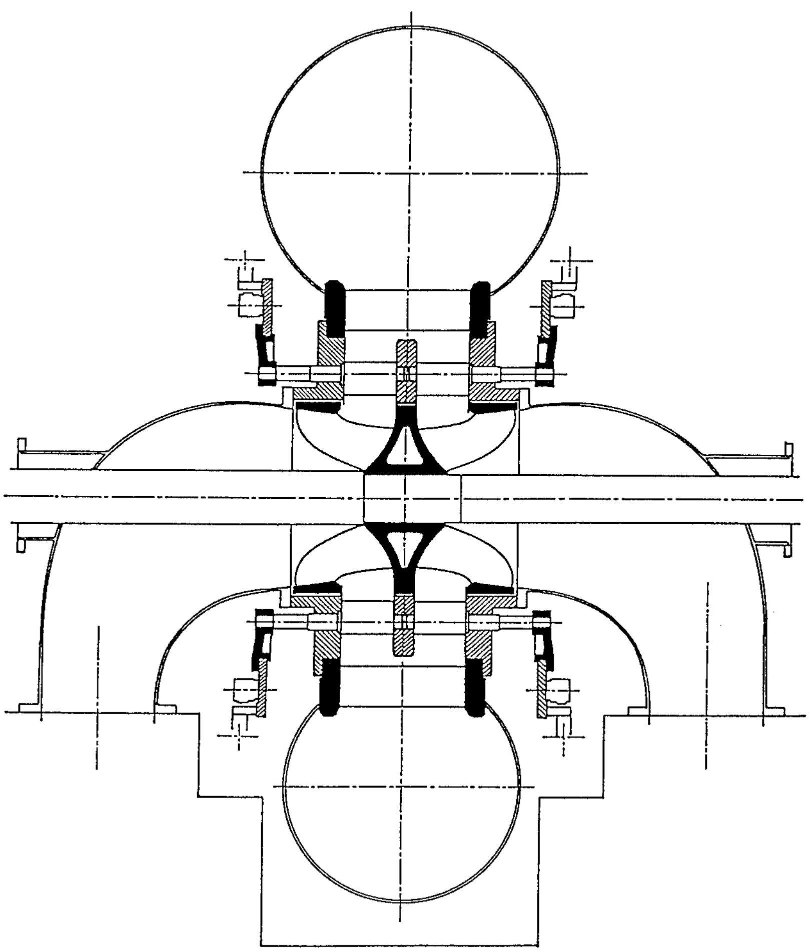


Fig. 3

The main advantage of such an arrangement is to have performances nearly equal to those of two individual units, with almost the same civil works and electrical equipment costs as for one turbine.

Due to the independent guide vanes systems, the turbine cost is around 15 % higher than the cost of a "normal" double Francis turbine unit.

Double Runner Francis Turbine with Ring Gate

For the operating heads over 40 m, the leakage thru the closed set of the independent guide vanes is significant when the double runner turbine is operating at low outputs. This leakage can be greatly reduced when a ring gate is used for shutting off one half of the double runner instead of the independently operated guide vanes.

The ring gate arrangement is shown on figure 4. The guide vane apparatus is the same as for the double runner Francis turbine, the ring gate closes one half of the runner when the flow drops below 1/2 of the rated discharge.

In this arrangement the intermediate ring is still there, its functions being hydraulically to insure proper flow pattern to the runner in use and mechanically as a seat for the ring gate seal.

The guide vanes are therefore still split into two halves and secured together by a threaded rod located in their shafts. The two guide bearings are located in the turbine covers and there are bushings at the intermediate ring.

This ring gate is installed in a fabricated housing welded on the stay-ring and the turbine cover on the side opposite to the guide apparatus operating ring.

The ring gate slides between the stay vanes and the guide vanes down to the intermediate ring. It is operated by three screw and nut drives located at 120° on the ring gate housing. The screw drives are controlled by an electrical actuator and are synchronized by a chain and gears.

The efficiencies of this arrangement are better at part loads than those of the independently controlled guide vane mechanism. This is because the leakage losses of the closed ring gate are much smaller than the leakage losses of the closed guide vanes. The runner crown seal leakage is the same in both cases.

Cost is slightly higher than the former solution, around 20 % more than a "normal" double Francis turbine.

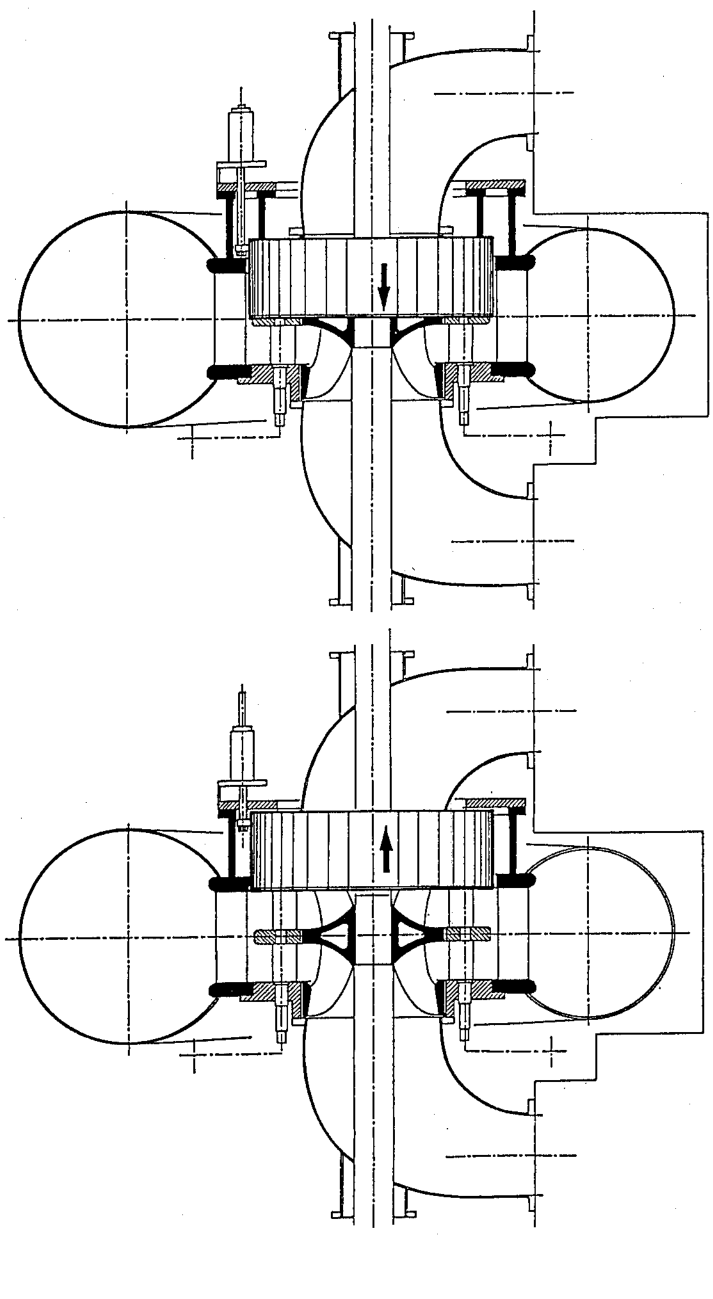


Fig. 4

DOUBLE FRANCIS TURBINE WITH RING GATE

Conclusion

Fig. 5 shows the main data of the double runner Francis turbine arrangements installed recently in North America. These examples demonstrate that the modern technology of the double runner Francis turbine can contribute significantly in finding economic solutions for small hydro sites with wide flow variations because it can operate very efficiently at part load.

When necessary part load performances may even be improved by combination of units of different sizes.

MAIN DATA OF RECENT NORTH AMERICAN PROJECTS WITH DOUBLE RUNNER FRANCIS TURBINES

YEAR	PROJECT	STATE	OWNER	ENGINEER	NO. OF UNITS	POWER kW	HEAD m	SPEED rpm	RUNNER dia. mm	TYPE
1986	SCARFE LAKE	ONTARIO	Scarfe Hydro	Walker	2	1190	21	600	740	
1987	WAPPINGERS	NEW YORK	Might		1 1	1035 1035	24 24	600 600	740 740	Double GV
1988	SERPENT RIVER	ONTARIO	SR Power Co	SNC	2	3203	45	600	840	
1989	BAILEY PONDEROSA	CALIFORNIA	Owl		1	1477	80	1200	430	Ring Gate
1990	BELLY RIVER	ALBERTA	Canadian Hydro Dev.		1	3100	45	720	820	
1990	BLACK RIVER	ONTARIO	Conwest Exploration	Acres	2 1	4650 4650	47 47	600 600	925 925	 Ring Gate
1991	WATERTON	ALBERTA	Canadian Hydro Dev.	CH2M Hill	1	2830	44	720	770	Ring Gate
1991	SAINT MARY	ALBERTA	Canadian Hydro Dev.	CH2M Hill	2	2410	53	900	650	Ring Gate
1992	MISEMA	ONTARIO	Misema Power Partnership	Sandwell Inc	1	3850	41	600	925	Ring Gate
1992	COULONGE	QUEBEC	Hydro Pontiac	Tecsult	2	8500	46	400	1290	

Fig.5

Sizing Turbines to Use Municipal Water Supply Flow

E. June Busse, P.E.[1]
Jerry Waugh, P.E.[2]
R. Joseph Bergquist, P.E.[3]

Abstract

The hydroelectric potential in a municipal water system can be successfully developed, provided that care is taken to design the facilities to work within the limits of the system. Water system reliability and a continuous water supply are foremost when serving a community. Selection of the proper type and size of equipment are fundamental to a successful project.

When sizing hydraulic turbines to optimally develop the energy potential in municipal water supply systems, an economic study must be performed to determine the turbine size and capacity to give the best economic performance over the life of the project. The range of turbine sizes to be studied should be defined by project economic performance, not by the rules-of-thumb typically used in prefeasibility studies.

This paper discusses one method that can be used to reliably select the size and type of hydroelectric equipment for installation in a municipal water supply system.

Introduction

For Boulder, Colorado, the generation of hydroelectricity is a by-product of the municipal water supply system. Like many other cities, Boulder's water supply

---

[1] Utilities Project Manager, City of Boulder, 1739 Broadway, Boulder, Colorado 80306.

[2] Senior Water Resource Engineer, Engineering Consultants, Inc., 5660 Greenwood Plaza Boulevard, Suite 500, Englewood, Colorado 80111.

[3] Senior Water Resources Engineer, Engineering Consultants, Inc.

originates in a mountain range. These types of water supply systems are designed to take advantage of gravity to deliver the water to the user. In Boulder's system, water is delivered to the water treatment plant, or to the municipal water distribution system, under very high pressure. This pressure must be reduced to atmosphere before the water can be treated, or reduced to operational pressures before the water can be used for domestic purposes.

The pressure has historically been wasted with pressure reducing valves. However, another available choice is to use the pressure to generate electricity. The hydro projects can be installed with no or minor environmental impact, because the dams, lakes, and pipelines are already in place for delivering the municipal water supply. This is the case with Boulder's water supply system.

Developing the hydroelectric potential in a municipal water system can be accomplished, provided that care is taken to design the facilities to work within the limits of the municipal water system. Water system reliability and a continuous water supply are foremost when serving a community. Sizing a municipal water supply hydro plant by traditional methods will result in equipment that is too small to deliver the peak water demands. This results in operational problems and a substantial loss of revenue.

Avoidable Operational Problems

Equipment selection is the critical element in the financial and operational success of a hydro facility in a municipal water supply system. The process for selection of the type and size of turbine, generator, and valves varies from the methods used for traditional hydro projects.

Traditionally, during reconnaissance level studies of a hydro site, a limited number of turbine capacities are studied to determine if the project has merit for further study. The range of turbine design flows studied for a traditional plant normally ranges from 15 to 30 percent exceedance on the flow duration curve. However, this rule-of-thumb is based on river hydrology. Rivers have much higher peak-to-average-flow ratios, but have a less reliable peak recurrence than a municipal water supply system. Using the rule-of-thumb to size a hydro plant in a municipal water system will result in equipment that is too small to deliver the peak water demands and a loss of substantial revenue. The additional reliable revenue from the municipal water system can be used to offset the incremental cost of the larger equipment. The 15 to 30 percent exceedance on the flow duration curve should be used only as the starting point for sizing of the equipment, not for final equipment selection.

Using the preliminary size selection based on the rule-of-thumb, the designer should evaluate how the facility will operate during water supply peak conditions

with equipment of this size. For example, if the selected hydroelectric equipment is able to pass 0.28 cms (10 cfs), but the summer peak is typically 0.42 cfs (15 cfs), the equipment cannot pass the peak flow. Since the equipment cannot pass the peak, the hydro equipment will either be shut down during peak periods or must be capable of operating in parallel with a bypass system.

Peak water flows in a municipal water system occur two to three months out of the year in the semi-arid climate of Colorado. These factors can substantially reduce the revenue for the facility. When the turbine and the bypass are both operating at peak water flows, the turbine cannot generate at full load. Since these turbines are typically supplied by long pipelines, increasing the pipeline flow increases the pipeline friction loss and lowers the head available to the turbine. The net result is that the turbine is operated at a lower head, flow rate, and efficiency; and less hydropower revenue is produced. Also, the potential energy of the flows passing through the bypass valve is dissipated and not put to beneficial use.

The method used to properly select the equipment size and type involves performing an economic analysis to refine the equipment sizing. This is one of the principal concepts that should be considered for any hydroelectric development in a water supply system.

Sizing of Turbine for the Silver Lake Project

The Silver Lake Pipeline conveys raw water from the City's high-mountain watershed to Lakewood Reservoir which serves as a regulating reservoir between the Silver Lake and the Lakewood Pipelines. Silver Lake Pipeline is 5.8 km (3.6 miles) long with an average slope of 7.9 percent and static head of 469 m (1,540 feet). The existing inlet-controlled Silver Lake Pipeline has a flow capacity of 0.85 cms (30 cfs). Pressure reduction in the inlet-controlled pipeline is presently accomplished with a series of internal pipe cascades, hydraulic jumps, and pipeline friction.

Replacement of the Silver Lake Pipeline is required for two reasons. The pipeline is in an advanced stage of deterioration due to age and the open channel flow characteristics of the pipeline force (entrain) air into the water. Air entrainment also occurs in the Lakewood Pipeline downstream. The enormous quantities of air entrained in the water cause severe operational problems in the water treatment process. While Silver Lake Pipeline is considered to contribute a modest portion of the air entrained in the water, the City decided for a number of reasons related to system operation to replace the pipeline with a pressurized, outlet-controlled pipeline.

The proposed replacement pipeline will have a nominal flow rate of 0.88 cms (30.9 cfs), and the same static head as the existing pipeline. The proposed Silver

Lake Hydroelectric Plant will be located at the outlet of the Silver Lake Pipeline and have a short tailrace to Lakewood Reservoir.

Since water is a precious resource in Boulder, the City operates the hydroelectric plants in a flow control mode. The configuration of the water system prohibits running additional water through the hydroelectric plants for additional electric generation. The amount of electricity produced is based solely on the amount of water flowing through the system to meet the municipal supply demands.

Before the turbine sizing analysis was started, the type of turbine was identified based on the turbine inlet pressure, flow, and the discharge pressure. The turbine flow and head operating ranges could then be determined. The range of operation was determined before energy generation calculations were made.

For the Silver Lake Project, the energy calculations were made using the flow duration curve method. The Silver Lake pipeline flow duration curve is a result of the City's direct flow water rights on North Boulder Creek and supplemental water storage releases from the reservoirs in the City's watershed. The water is diverted into the pipeline to meet the City's water demand.

The flow duration curve shown on Figure 1 represents the flow record for the raw water demand of 1990 that could be supplied through the Silver Lake Pipeline. Since no additional water is available from this supply, additional future flows have not been included in this analysis. The water demand flow duration curve for the Silver Lake pipeline is not at all comparable to the typical unregulated flow duration curves for traditional sites.

Because of the different shape of the flow duration curve, the best turbine size may not fall in the bounds established using the 15 to 30 percent exceedance rule-of-thumb often used at the reconnaissance level. For the water demand flow duration curve, 15 to 30 percent exceedance corresponds to flow rates of 0.76 to 0.72 cms (27 to 25.5 cfs), respectively, a very narrow range. At the feasibility level of study, an economic analysis is necessary to test turbine capacities outside the bounds established in previous prefeasibility studies.

Since the Silver Lake site has 469 m (1,540 feet) of static head (some of which will be lost due to pipe friction over the 5.8 km (3.6 mile) length of the line), an impulse turbine was selected for the study. In addition, the discharge of the water into a tailrace upstream of Lakewood Reservoir is compatible with the outlet pressure requirements of an impulse turbine. A multiple jet impulse turbine was considered ideal for the Silver Lake Pipeline hydrostation due to the turbine's high efficiency over a wide range of flows.

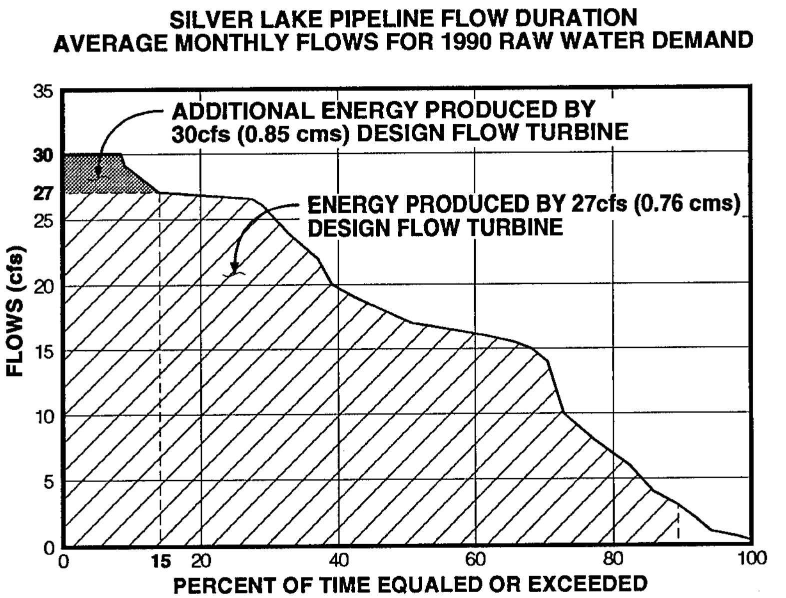


FIGURE 1

The previously prepared screening and reconnaissance level studies were examined as a first step in the turbine sizing process. A detailed economic analysis was carried out during this study to determine the optimal turbine and supply pipeline size for the Silver Lake Project. Computer spreadsheets were developed to determine the power and energy produced by a turbine with a selected flow rate and pipe size. The construction cost was estimated based on the turbine flow rate and pipe size selected, and the economic performance of the project. Turbine design flow rates ranging from 0.64 cms (22.5 cfs) (73 percent of pipeline nominal water supply design flow) to 0.85 cms (30 cfs) (100 percent of pipeline nominal water supply design flow) were evaluated. The pipe sizes studied for each turbine design flow rate ranged from 610 mm (24 inches) (the minimum size to provide a maximum flow velocity of 3 mps (10 fps) at the water supply nominal flow rate) to 965 mm (38 inches) (a size estimated to be well above the economic size).

For the water demand flow duration curve, the spreadsheet analysis showed that the most economic project consists of a 610 mm (24-inch) diameter pipeline, a 0.85 cms (30 cfs) turbine and a 2.75 megawatt generator. The turbine flow rate found to be optimum by the economic analysis was larger than the 0.76 cms (27 cfs) maximum turbine flow rate indicated by the 15 to 30 percent rule of thumb used at reconnaissance level studies. The optimum turbine flow rate also nearly equals the nominal water supply capacity of the pipeline. The hydraulic data and economic performance parameters for these two flow rates are compared in Table 1.

TABLE 1

HYDRAULIC DATA AND ECONOMIC PERFORMANCE PARAMETERS

Turbine Design Flow, cms (cfs)	0.76 [1]	(27)	0.85 [2]	(30)
Pipe Diamete, mm (inches)	610	(24)	610	(24)
Pipe Velocity, mps (fps)	2.6	(8.6)	2.9	(9.5)
Static Head, m (ft)	468	(1,537)	468	(1,537)
Pipeline Friction, m (ft)	57	(187)	69	(228)
Net Head, m (ft)	411	(1,350)	399	(1,309)
Generator Capacity, kW	2,550		2,750	
Annual Energy, GWhr	13.4		13.6	
Benefit Cost Ratio	1.06		1.07	
Present Worth of Net Benefit, $	483,000		552,000	

[1] Design flow rate corresponding to 15 percent exceedance on the water demand flow duration curve.

[2] Design flow rate with best economic performance when using the water demand flow duration curve.

Other Considerations

Even with the proper selection of the turbine size, flow conditions occur which require parallel operation of the bypass valve. The turbine bypass valve should be selected to match the flow characteristics and pressure-reducing capabilities of the turbine. Mismatching the turbine and the bypass valve may result in the bypass valve taking the majority of the flow and "starving" the turbine. The turbine generator output will potentially drop below the minimum acceptable output and the system controls will shut down the turbine generator. This results

in a considerable loss of revenue during a peak power generation period. By installing a bypass valve with compatible flow control characteristics to the turbine, hydropower generation during peak flows is possible. Passing flows through the turbine bypass pressure reducing valves for an extended period of time also shortens the life of the internal components of the valve.

Selection of the proper turbine type also can be critical for reliable operation of the water system. Two turbine types have been installed in Boulder's water supply system. Reaction turbines are excellent for in-line applications and impulse turbines discharge the water at atmospheric pressure.

When a turbine is supplied by a long distribution pipeline and feeds a distribution grid, the possibility of a high pressure rise upstream and severe pressure drop and water column separation downstream are dependent upon the amount of flow the turbine chokes at runaway speed. For reaction turbines with a specific speed selected to give the best efficiency, a substantial flow choking occurs at runaway speed. However, reaction turbines have been developed with higher specific speeds and slightly less efficiencies that only choke the flow 10 percent at runaway speed. These units have been successfully used to eliminate surge problems due to sudden loss of load and should be considered when specifying a turbine for a municipal water system with long upstream or downstream pipelines, or where water hammer could cause serious problems.

In municipal water systems, water is delivered to the customer at a pressure varying from 276 to 1,034 KPa (40 to 150 psi), depending on the specific site requirements. Since impulse turbines discharge the water at atmospheric pressure, this type of turbine cannot be installed within the water distribution grid where the turbine discharge pressure must maintain a positive pressure. However, the flow choking situation experienced with reaction turbines can be avoided with impulse turbines equipped with flow deflectors for each needle valve.

An undersized turbine results in less capacity and energy revenue than potentially available. The capacity payment for Boulder's projects, can be as high as 70 percent of revenue for the life of the project.

Conclusions

Sizing hydraulic turbines to optimally develop the energy potential in municipal water supply systems requires completion of an economic study. The range of turbine sizes to be studied must be defined by project economic performance, not by the rules-of-thumb typically used in prefeasibility studies.

Sizing the water supply system hydro facilities large enough to pass the peaking flow of the system provides several benefits to the owner. These benefits are:

(1) The revenue will be larger due to the ability to operate during high flow times.
(2) The system will be more reliable for water supply operations.
(3) The cost to procure and install the larger equipment will normally be recovered quickly due to increased revenue.

Boulder has demonstrated that hydroelectric facilities can successfully be installed and operated in a municipal water supply system. The renewable energy developed by the water delivery operations has been harnessed for a benefit to the environment, retained the energy revenues in the local economy, and has provided another source of revenue for the City.

YACYRETA - WESTERN WORLD'S LARGEST KAPLAN TURBINES

Randy V. Seifarth *

Abstract

The Yacyreta Project on the Parana River, currently under active construction, is the largest hydroelectric development underway in the world. The design of the large Kaplan turbines presented significant challenges to the hydraulic and mechanical engineers, leading to new high levels of performance and interesting solutions in the layout, design, manufacturing and installation of the principal turbine components. These challenges and the solutions, therefore, are described in this paper.

Introduction

The Yacyreta Hydroelectric Project will house the largest diameter Kaplan turbines in the Western Hemisphere and has offered special challenges to the designer. These challenges were identified during the proposal and defined as design targets after contract award according to the list below:

- Optimization of turbine hydraulic performance
- Define deflection and fatigue limits for major components
- Maximize reliability and serviceability of turbine components
- Design components to anticipate and minimize manufacturing errors
- Adapt the designs for flexibility to be manufactured worldwide
- Optimize material and manufacturing costs
- Coordination of the design with site installation requirements

* Manager, Product Engineering, Voith Hydro, Inc., USA, P.O.Box 712, York, Pennsylvania 17405

This paper will discuss the integration of these targets with design features and manufacturing processes utilized by Voith for the Yacyreta Project.

The Yacyreta Hydroelectric Project is a joint development between the countries of Argentina and Paraguay. The Owner is the Entidad Binacional Yacyreta (EBY), responsible to oversee and administer all project works and to operate the power plant after commissioning. The civil and plant engineering is contracted to CIDY, an international consortium of consulting engineers.

The Project is located on the Parana River between Argentina and Paraguay downstream of the Itaipu Power Plant. The overall scope of the project includes 64 kilometers of dikes and earth dams, 2 sets of spillway gates, a navigation lock, irrigation outlets, fish passageway, traffic causeway, powerhouse, transmission/distribution facilities, and 20 Kaplan turbines.

The turbine contract, consisting of 20 turbines, was awarded to Consorcio Voith and General Electric Canada in March 1988, after a long period of negotiation, based on the offer made in 1980 by Voith's predecessor in the USA, Allis-Chalmers. The hydraulic design was assigned to Voith, responsible for 100% of the hydraulic geometry and model testing. The mechanical design is split between Voith and General Electric Canada and manufacturing takes place in the USA, Canada, Argentina, and Paraguay. Installation of the turbines will be completed by Paraguay under the direction of Consorcio Voith and General Electric Canada Field Supervision team. This project represents a massive undertaking as demonstrated by the plant data shown in Table 1.

TABLE 1 - Plant Data - Yacyreta

Number of units	20
Type of units	5 blade vertical Kaplan
Rated unit capacity	138 MW
Maximum unit capacity	160 MW
Rated head	21.3 m
Head range	19.5 - 24.1 m
Synchronous speed	71.4 rpm
Rated flow	704 cms
Turbine contract award	04-Mar-88
Commercial operation - Unit 1	1994
Commercial operation - Unit 20	1997

The hydraulic design is based on state-of-the-art technology and includes full model testing which was completed in June 1988 at the Voith Laboratory in the USA. The testing resulted in optimization of the spiral case, wicket gates, stay vanes, Kaplan blades, and draft tube to achieve +95% maximum prototype efficiency as shown in Figure 1.

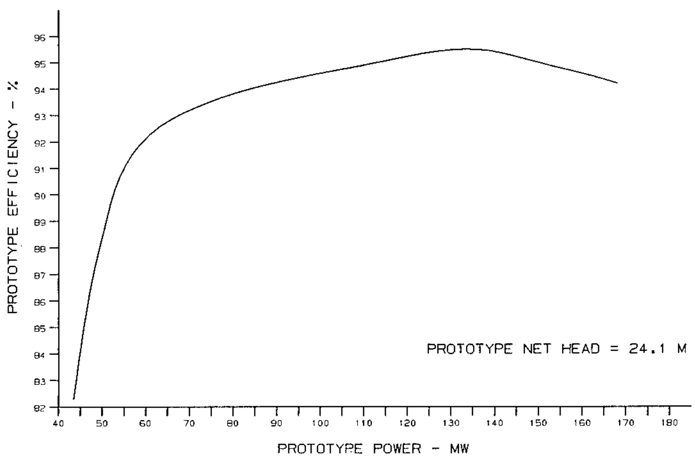


Figure 1 - Prototype Efficiency Curve

Each wetted component was carefully designed to obtain the best efficiency while insuring manufacturability. The stay vane inlet angles were set to match the flow angles and the discharge angles to provide zero degrees incidence angle to the wicket gates for peak efficiency at the rated point. All stay vane shapes were carefully reviewed for ease of manufacture. The trailing edge shape of the wicket gate was selected for best hydraulic performance and adapted well into the fabricated design. Flow losses in the semi-spiral case were minimized by developing a large cross-section with low velocities and a relatively uniform height-to-width ratio. The runner hub shape was optimized hydraulically by limiting the sphere to 42.8% of the blade diameter and minimizing the size of the upper cylinder diameter, presenting mechanical challenges that will be addressed later. Optimization of the blade shape is the single feature that required the most testing lab work and is the result of iterative modification during the model testing that resulted in outstanding performance. The draft tube performance likewise, was the result of progressive fine-tuning, leading to maximum pressure recovery while remaining relatively simple for construction of concrete form work. The model testing included measurement of gate torques, blade torques, cavitation characteristics, and axial thrust, in addition to the standard

model performance data. The timely analysis and preparation of this data permitted a smooth transition to the mechanical design.

The mechanical design of these giant machines required consideration of component weight limits, restricted availability of suppliers of large castings, very large deflections of components under operating loads, wide variety of manufacturing methods, and the need to consider interaction of assembled components. These influences helped to define the current design.

The embedded components of the turbine consist of the draft tube liner, discharge ring, bottom ring, stay ring and pit liner as shown on the turbine cross-section in Figure 2.

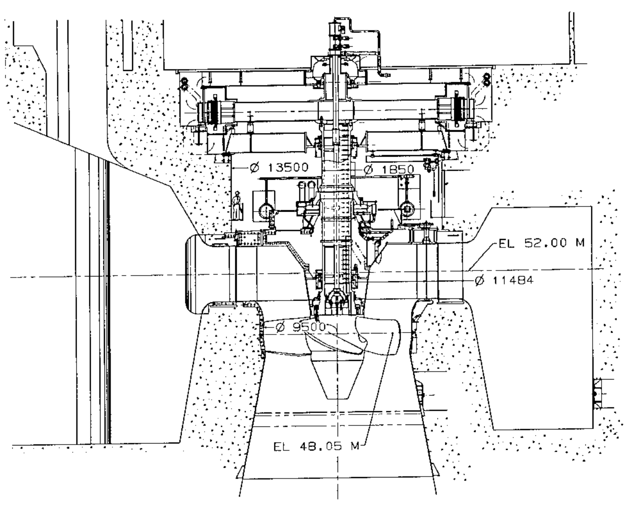


Figure 2 - Turbine Cross-Section

The unit is designed for embedment of the discharge ring and bottom ring last, after the stay ring and pit liner are embedded, thus allowing fine-adjustment of the centerline of the turbine at the final stage of embedded parts. The discharge ring and bottom ring are joined by a flange at the horizontal centerline of the blades. The discharge ring is lined with 18% chromium and 8% nickel composition stainless steel plate for 1100 mm below the unit centerline.

The rotating parts include the Kaplan runner hub assembly and the turbine shaft. The 5 adjustable pitch runner blades, with integral trunnions and anticavitation fins on the peripheral edges, are stainless steel castings of 13% chromium and 4% nickel composition. All wetted surfaces of the blades are CNC machined to provide the highest possible hydraulic performance and consistency from blade to blade. The blades are retained with the rocker arms by use of expanding dowels through the blade trunnion. The Voith runner hub is fabricated using formed steel plates, up to 305 mm thick, for the shell, forged rings at the trunnion openings, and a cast carbon steel inner hub as shown in Figure 3. The weld geometry for all hub assemblies were premachined for proper fit-up of the weld areas. The sections were welded together using a narrow gap submerged arc welding technique.

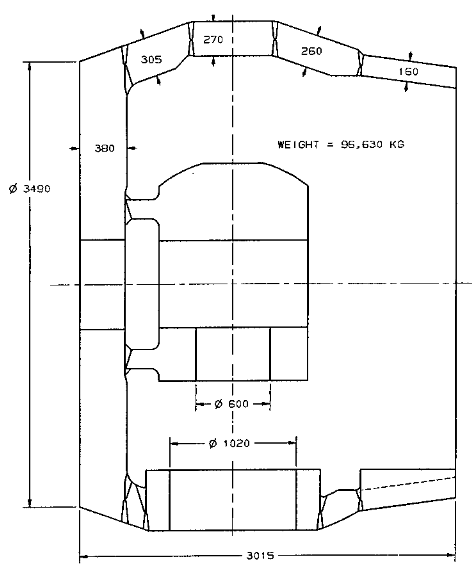


Figure 3 - Runner Hub Fabrication

Figure 4 shows the arrangement in the fabrication shop. The advantages of this design are improved quality at lower cost and reduced lead times for the materials, as casting the hub in one piece would be somewhat impractical.

The moving parts in the Kaplan runner assembly are required to be fully independent of the deflector, resulting in an unusually long and heavy

runner hub to support the anti-rotation key for the crosshead in the blade operating mechanism. The assembly became more difficult but the extra material improved the hub stiffness and reduced deflection under operating load. The servomotor is located in the turbine shaft at the juncture with the hub. This design combines the massive hub and shaft to form a servomotor cylinder.

The internal operating mechanism of the hub is designed for normal operation under fatigue loading and temporary overloads at full runaway speed. At runaway speed condition, the radial deflection of the hub and blades would cause binding of the operating mechanism in the linkage bushings if conventional straight bushings were used. The impact of the effects of deflection are eliminated by using spherical linkage bearings.

The shaft is manufactured from hot-rolled plate steel, 170 mm thick, to form cylinders and ring forgings using a narrow gap, dual tandem submerged arc welding technique to minimize deposited weld volume and insure high deposition rates. The quality of these welds is checked by radiographic and magnetic particle inspection.

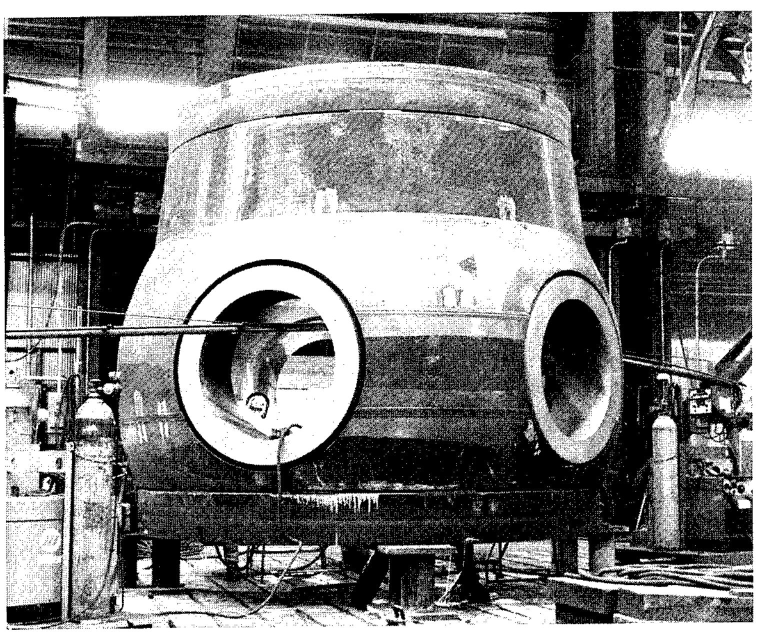


Figure 4 - Runner Hub in Shop

The hydraulic thrust and torque cause significant deflection of the runner hub, especially at runaway speed. The radial deflection is 0.63 mm and torque wind-up is 2.89 mm at the runner centerline. For this reason the shaft, hub, and deflector were designed to combine the stiffness of each to reduce the deflections and minimize binding of the blade trunnion bushings - see Figure 5.

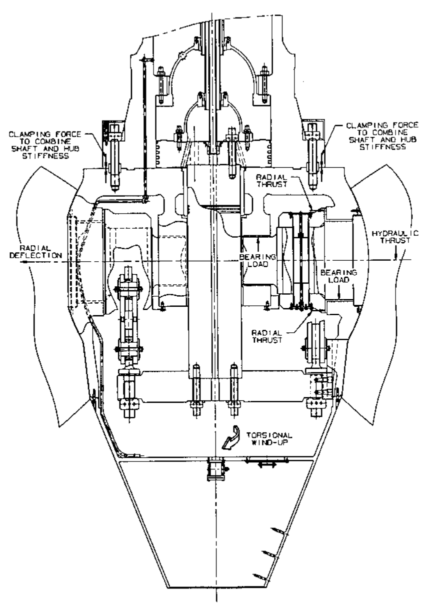


Figure 5 - Kaplan Runner Assembly

The finite element analysis requires that the shaft and the hub be integrally clamped together using prestressed studs and hollow drive dowels instead of the conventional fitted bolts. The required clamping force is 7260 metric tons shared by 30 stud bolts that are M110 x 6 diameter. Benefits of using studs instead of fitted bolts are improved access during assembly to compress the piston rings on the blade servomotor, ease of applying large prestress force (using stud tensioners), and elimination of through holes in the runner hub flange to reduce risk of oil leakage from the hub.

The rotating parts are supported by a thrust bearing mounted on the head cover. The thrust bearing support ring, thrust ring, and thrust block are not split so that maximum stiffness is obtained, insuring uniform load distribution, and preventing offsets at the split faces. The thrust bearing shoes are equipped with a high pressure lift system. The total thrust load at normal operation is 2500 metric tons at a unit pressure of 3.82 MPa.

The shaft is guided by the turbine bearing mounted in the inner head cover and the intermediate guide bearing which is mounted on a support bridge directly below the generator. All bearings are externally cooled by oil-to-water heat exchangers with cooled and filtered oil returned to the oil bath using a pressure lubrication system fed to the space between the shoes. The bearings are designed to operate for 30 minutes at full runaway speed.

Water control is achieved by 24 wicket gates supported by conventional grease lubricated bronze bearings mounted in the outer head cover and bottom ring. The outer head cover is supplied with a compressible rubber strip and the bottom ring is supplied with an inflatable rubber strip to seal the ends of the wicket gates. The wicket gate seals minimize the amount of head water lost to leakage.

The wicket gates are designed to optimize the manufacturing process. The hydraulic and mechanical requirements permitted the gates to be made from formed plate steel with standard bar shapes for the leading and trailing edges. The stems are forged steel and the end plates are stainless steel. All parts were designed to be assembled and welded in manufacturing fixtures that creates an assembly line appearance during fabrication.

The wicket gates are operated by the conventional linkage, gate ring, and gate servomotor arrangement. The operating system is supplied with shear pins for overload protection and a gate restraining device to brake gate movement after shear pin failure. Head pressure is sealed from the turbine pit by a standard packing box seal equipped with four rings of compressible packing.

The turbine components for Yacyreta can be classified into four groups of equipment; Embedded; Stationary or Distributor; Rotating; Control. The order of this listing represents the general sequence of site installation.

The embedded components make extensive use of internal spiders and external anchor rods to obtain the proper roundness and concentricity during concreting operations. Key inspections are required before, during and after concreting to insure that correct tolerances are maintained. As mentioned earlier, the bottom ring-discharge ring assembly is the final major components to be embedded. This assembly is independently centered to

the unit centerline, plumbed and set to correct elevation relative to the stay ring flange (which supports the head cover), prior to embedment. In the field, roundness of the discharge ring bore is difficult to achieve especially considering that this component is designed in four sections, fabricated from 18-8 stainless steel, and easily distorted from most handling during shipment. On this basis, heavy bracing is factory installed in the bore after final machining but prior to removal from the machining table, to maintain the bore size. The final adjustment is obtained by site grinding of the stay ring flange which supports the outer head cover, to assure proper water passage opening between the bottom ring and outer head cover.

The stationary components must be aligned to the unit centerline established by the discharge ring-bottom ring. This is accomplished by installation of the wicket gates, installation of the outer head cover and centering, then inspecting free operation of the wicket gates. The outer head cover is dowelled to the stay ring after all inspections are completed.

The site assembly and installation of the rotating components requires extensive use of special fixtures, tools and lifting equipment. Erection plates and support stands are used to assemble the Kaplan runner from underneath. All fasteners M48 and larger are designed to be site tightened using stud tensioners so that reliable prestress values are obtained. The internal runner components and blades are installed, followed by the shaft, inner head cover, packing box, turbine guide bearing support, intermediate head cover, and gate operating ring. This assembly is 584 metric tons and 17.7 meters high (see Figure 6) and is lifted by linking four 165 metric ton capacity bridge crane hooks using a combination of lifting beams. After installation of the runner deflector cone the assembly is lowered on to the outer head cover.

The thrust bearing is supported on the head cover and is assembled next. Proper alignment (perpendicularity and elevation) of the thrust bearing with the rotating parts is accomplished by a site machined shim plate installed between the head cover and thrust bearing support prior to installation of the generator rotor. The remaining parts are installed in the turbine pit followed by the thrust bearing shoes and loading of the thrust bearing.

The intermediate guide bearing support serves multiple tasks including pit monorail support, generator air housing cover, generator brake-jack support and air baffle support. The support is fully assembled on the powerhouse floor with its foundation plates and installed in blockouts at the top of the pit liner. It is adjusted for elevation and center then grouted in place. Generator work is finalized at this time.

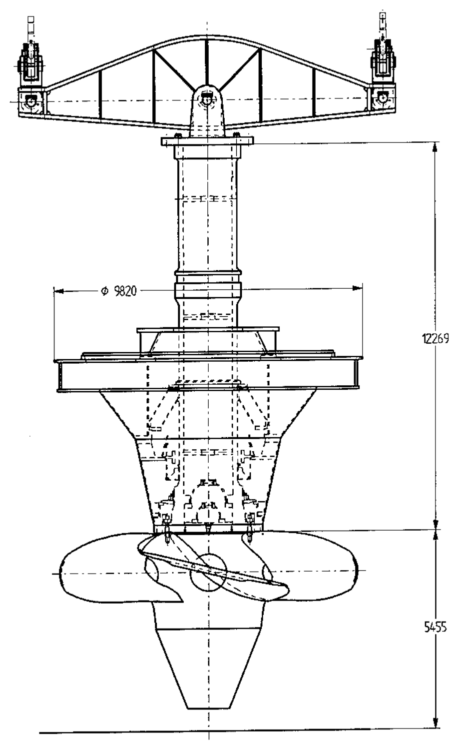


Figure 6 - Field Lifting Arrangement

The Kaplan oil pipes and oil head can now be installed while work proceeds for the governor, turbine control cubicle, motor control center, and interconnecting piping and wiring. Currently, the site work is beginning for the Kaplan runner assembly and the installation tools are being used for the first time. These trials will certainly bring some modifications and improvements for the remaining units which the turbine designers anticipate and welcome.

Yacyreta represents the transition of old methods to new methods where all design work is completed on CAD systems, turbine components are CNC machined, and international teams participate in all phases of the design, manufacture, and installation. The combination of diverse teams provides a collection of the best ideas at one project site, a result that is very beneficial to the owner.

VARIABLE SPEED POWER CONVERSION TECHNOLOGY FOR HYDRO APPLICATIONS UNDER 10 MW

VAN A. JOHNSON, C.E., M.B.A.[1]
ALEKSANDER S. ROUDNEV, Ph. D., M.E.[2]
HANS MEYER, A.E.[3]

In the early days of American electrification with an abundance of water sources and little demand for power, there was a tendency to be very conservative in the design of hydro electric plants. Engineers would design for minimum anticipated flows to assure constant power station operation. A substantial amount of water was lost over the dam or at the diversion point for much or all of the year.

Recently, with great increases in demand, engineers are constantly searching for more ways to produce energy. One way has been to modify power generation equipment to make it more efficient. With equipment efficiencies now reaching into the mid ninety percent, it might be said that we have reached a limit on what can be done physically to increase power output through design.

Another way to produce more energy is to efficiently capture more of the flows of a given source. This effort included a study of better equipment sizing and planning. Limits were also established as capital costs for equipment are balanced with increasingly marginal income from power output. However, if the generator efficient operational RPM range could be expanded, the same traditional turbine could produce more energy without increasing capital costs.

---

[1]Vice President-Operations, Trillium Energies, SLC, UT.

[2]Senior Project Engineer, BGA International, SLC, UT.

[3]President, Omnion Power Engineering Corp., E. Troy, WI.

This conclusion was also reached by hydro industry leaders at the North American Hydroelectric Research and Development Forum held in February 1992 in Kansas City, Missouri. These leaders cited the following recommendation: "An essential research project in the hydro equipment area is exploration of variable speed operation of turbine-generators. To date, all commercial U.S. hydroturbine-generator units rotate at constant (synchronous) speeds. If this rotation could be varied, the turbine's operation could be converted into power more efficiently - thus generating more energy."

Variable speed operation had actually been commercially demonstrated by the time of this conference. A 120 KW plant using variable speed technology was placed on line by Trillium Energies Inc. in December 1991 in Logan, Utah. This plant was built to capture the variable (30-90 CFS) flows that would be seasonably available at the required fishery bypass of a diversion dam. This variable speed power conversion system (PCS) allows the turbine and generator speed to be matched to the available flows thereby operating more efficiently and generating greater amounts of energy.

Variable speed power conversion technology has been under development for over 20 years. The Logan hydro electric plant is a product of state-of-the-art power electronic technology that has been applied in a wide variety of industrial and renewable energy applications including wind, photo voltaic, fuel cell, battery and hydropower.

For hydropower applications, the PCS is a three phase power conversion system using advanced insulated gate bipolar transistors (IGBTs) to convert variable frequency AC power from a rotary generator to 60 Hz utility compatible AC power. This conversion is accomplished in two steps: from variable frequency AC power to DC power and then back to constant frequency grid compatible AC power in real time. The PCS is designed to convert the maximum power available from the generator for the particular flow conditions and to deliver that power in synchronism with the operator's utility service.

The PCS controls are micro-processor based with sophisticated self-diagnostics. Performance and operating status are continually updated on local and remote control panels. Maximum power tracking is a feature that ensures peak power output at all times. The PCS automatically seeks peak power output by constantly increasing and decreasing current draw from the generator, thus slightly changing the turbine RPM to check for the highest possible power output under the

existing operating conditions. Changes in head and flow as well as specific gravity (in the case of slurry applications) are thus automatically translated into peak power output. High speed switching enables the PCS to deliver a sinusoidal output waveform with much less than 5% current harmonic distortion.

VARIABLE SPEED POWER CONVERSION

In order to better understand this conversion process an understanding of inductance is necessary. An inductor has the characteristic that current (quantity of electrons) flowing in it cannot be instantly changed. To equate this process to simple terms, an inductor can be thought of as a water wheel. **Figure 1** is an example of a water wheel operating like an inductor. In this figure, the water must flow through the water wheel, and in order to get to Tank 2, the water wheel must turn.

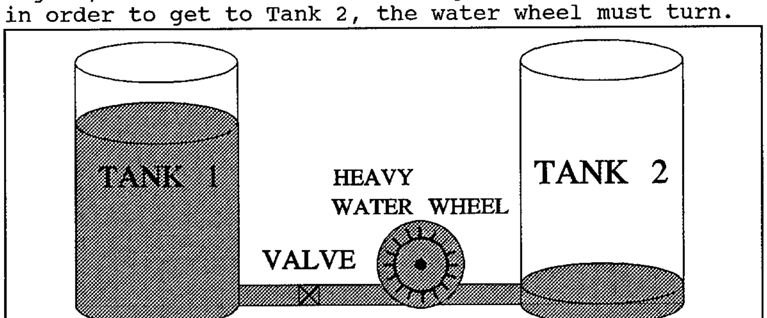


FIGURE 1

When the valve is first opened, the water begins to rotate the wheel. The flow is limited by the speed of the wheel. As the first tank continues to empty, the wheel turns faster until the two tanks are at the same level at which point the spinning wheel will continue to turn due to its inertia, pulling water out of Tank 1 and pushing it into Tank 2. At the point when the tanks were at the same level and the wheel was spinning, energy stored in the wheel was used to "pump" water from Tank 1 to Tank 2. This is similar to the energy storage capability of an inductor, except that the energy is stored in a magnetic field rather than in a rotating mass.

The PCS converter includes a bi-polar DC bus: a positive DC half and a negative DC half. To create AC current, an inductor and two switches are connected between the two halves of the DC bus and the AC service. To shape the

current, each half of the DC bus is alternately connected through the inductor to the utility. In that the DC voltage is higher than the AC voltage, current flows from the array to the utility. This is the same as Tank 1 being higher than Tank 2 in **Figure 1.** This process is represented in **Figure 2.**

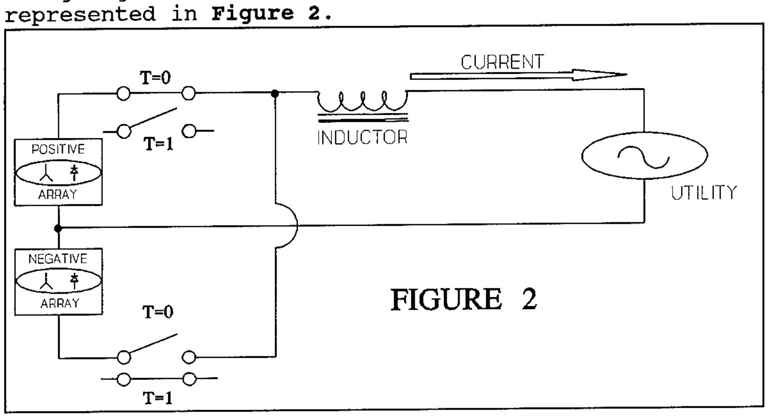


FIGURE 2

Eventually, the current flowing to the utility becomes greater than desired, so Switch 1 is opened and Switch 2 is closed. This happens at T=1 as shown in **Figure 3.** Since current is flowing in the inductor, the current slows down. This can be seen in the graph in **Figure 3.** This sequence is performed by the PCS at approximately 6000 times a second to give an AC current waveform, perfectly compatible with grid power, as represented by **Figure 4.**

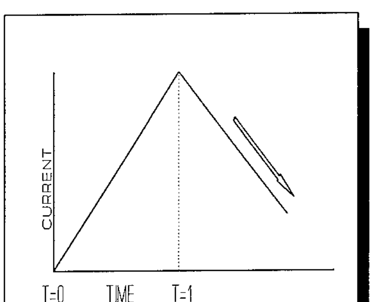


FIGURE 3

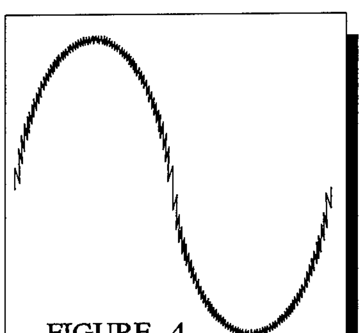


FIGURE 4

FLOW CONTROL

In order to control the frequency output in a traditional reactive turbine, a variety of mechanical or electrical flow control devices are used: wicket gates, ball-governor, passive resistors etc. With a variable frequency AC-DC-AC power conversion system such controls can be eliminated. This is best understood if the mechanical torque of a Francis or Kaplan type turbine is reversed making these units pumps. When a pump starts up from static conditions, the total current draw from the utility peaks at roughly 10 times normal operating load current level. This load spike is necessary to overcome the inertia of the pump mechanism and the system fluids. See **Figure 5**. There is a direct correlation, within certain ranges, of the amount of power required and the speed (RPM) of the pump.

If this system is now reversed into a Francis turbine-generator set, it is also true that there is a direct correlation between power output and speed of the turbine. **Figure 6** shows the relationship between current output and resultant changes in torque or flow of the turbine. As the current is momentarily increased, the turbine torque increases slowing the flow (Time=T1). As the flows decrease the current output decreases to equilibrium (Time=T2). To increase flows and resultant current output the current draw is momentarily decreased. All of these control functions can be programmed into the PCS which can take signals directly from a downstream pressure transducer of a pipeline, such as in a municipal water line, to maintain a required service pressure. Testing is typically required to identify the limits of flow control on a given set of conditions and the overall effect on the plant capacity factor, cavitation, etc.

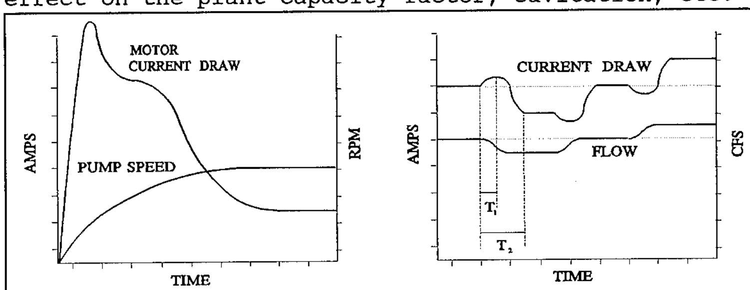
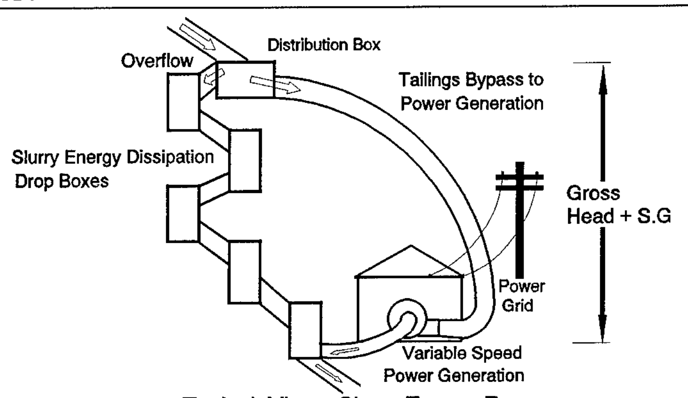


FIGURE 5 FIGURE 6

SLURRY PIPELINE ENERGY RECOVERY

At present, many facilities use gravity fed slurry pipelines to carry solid material. A good example of this is copper mines. Located mostly in mountainous zones, these mining operations face the need to dissipate excess energy of slurry being transported down the hill from concentrators to disposal sites, shipping ports or ore processing facilities. The possibility of capturing this excess energy, using reversed slurry pumps as recovery turbines, is economically attractive.

One thing that helps makes this so attractive is the direct increase in power output caused by the suspended solids in the slurry. Slurry specific gravities typically range from 1.2 to 1.6 and even higher. A slurry with a S.G. of 1.4 will have a power output potential of 1.4 times the same clear water flow and head. A second reason is that power is consumed on site, eliminating the need for power sales and wheeling. **Figure 7** is an elevation view of an energy recovery bypass around a series of slurry energy dissipation drop boxes.



Typical View - Slurry Energy Recovery
FIGURE 7

One of the main problems to overcome in this application is the narrow RPM range of effective operation of the fixed geometry hydro machine (slurry pump operating in reverse). Slurry flows from typical mining operations are subject to many changes. In any given day, the amount of ore or tailings transported can vary over a wide range. The specific gravity of the slurry will vary according to the type of materials and its concentration.

These conditions, plus changing flows, typically require a means of control for proper power production. Wicket gates or throttles, however, do not have sufficient wear life operating in slurry. Storage is out of the question as suspended solids will settle quickly and this operation must not interfere with the mining processes.

A variable speed power conversion system overcomes this narrow RPM range problem by allowing the slurry turbine to greatly expand the efficient range of power production. Rather than go off-line when RPM limits are exceeded due to the changing conditions described above, the variable speed system adjusts power output to exactly match the instantaneous head, flow and S.G. conditions. The PCS is programmed to constantly seek the optimum power output.

SLURRY PUMPS AS TURBINES, AN EFFICIENCY STUDY.

Slurry pumps operating as turbines, during water tests, consistently show a certain drop in efficiency as compared with pump performances. This trend, based on BGA International's test results of a limited number of units within the range of specific speeds below 2000, is represented by **Figure 8**. As can be seen, the efficiency is found to be 3 to 5% less for a pump operating as a turbine.

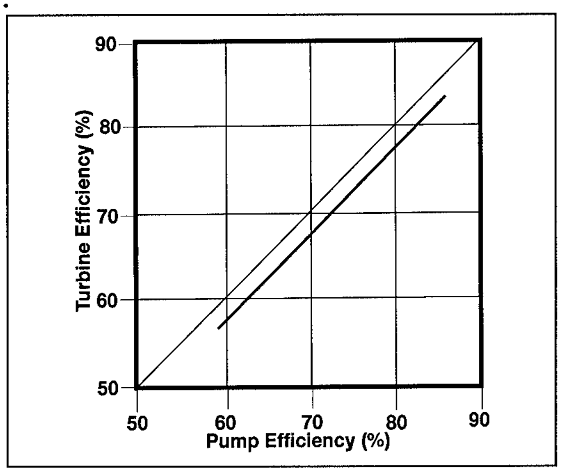


FIGURE 8

This difference will most probably exist every time a s lurry pump is used as a turbine, due to considerable

entrance losses through the turbine impeller, which has relatively thick vanes and uneven slurry flow distribution in the volute, since no guide vanes can be used. However, this efficiency drop can be reduced by optimization of hydraulic design of the impeller when used for turbine applications.

Thus there are two ways to optimize the capture of power from slurry energy recovery turbine installations using a reverse running slurry pump:

1. To use the slurry pump with the highest rated efficiency and with impeller modified for reverse flow.
2. To utilize variable speed power generation equipment to capture all variations in flows, head and specific gravity.

Slurry pumps running in reverse as turbines have typically flat efficiency curves around the best efficiency point (BEP). This feature, combined with high BEP and variable speed conversion system, can significantly increase the overall power output.

Figure 9 shows the BGA 4X3 test slurry turbine

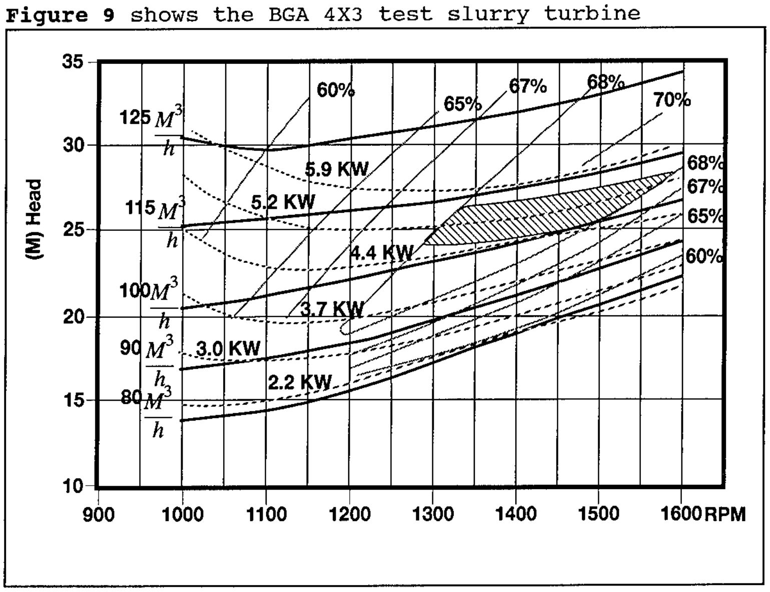


FIGURE 9

performance curve in head - RPM coordinates with lines of constant flows, power and efficiency. Referring to the shaded area, it can be seen that with a variable PCS, the power generating unit can accommodate 15% variation of head and 10% variation of flow, staying within the range of high efficiency (2% or less decrease in BEP) and certain variation of power output. Much greater variations in head and flow can be tolerated and only result in marginal efficiency losses but still allow the plant to continue to produce power.

Some special questions of slurry turbine operation, such as performance correction and wear life, are not discussed in this paper. It is worth mentioning that judging upon BGA's experience, slurry turbine wear life can be predicted based on available data of pump operation. In general, if the turbine runs within the head and flow limits of the same size pump, equal or lower wear can be expected.

A major benefit in using a pre-engineered pump as a turbine is reduced costs when compared to a custom engineered product. This can cut turbine costs 50 to 75% or more and shorten procurement time. However, adequate maintenance funds must be set aside for on-going liner and impeller replacement.

CONCLUSION

Variable speed power conversion technology is a viable consideration when looking at the options of power generation. While, not a panacea for all hydro sites, in those locations where head and flow conditions vary substantially, a variable speed application may generate more energy than conventional methods. One completely new area to consider is slurry line power recovery. The ability to eliminate traditional flow control devices as well as being able to handle changes in flow, virtual head and specific gravity makes this an ideal application. A very attractive feature of slurry energy recovery is that there is no requirement for licensing under FERC regulations.

Another application that can now be seriously considered is capturing power from municipal water pressure reduction valve (PRV) stations. In mountainous regions of the United States, pressure regulation of the municipal water systems is a necessity and a constant maintenance problem. Some supply lines investigated to date, are very large with a power generating capacity of 5 MW or more. These PRV applications have no capacity for traditional storage and have constantly changing flow

conditions. Variable speed technology is ideal for these applications.

Power from tidal action has been a fascinating topic for many years. There is only a small amount of head to capture but an almost unlimited flow makes tidal projects very attractive. One of the problems has been the constant changing head and flow conditions. As has been demonstrated by BGA in their slurry tests, (Figure 9) the variable speed technology has greatly extended the ranges of efficient power production. Changes in flows are almost instantaneously compensated for through a full range of RPM. As long as there is sufficient tidal action to turn the turbine impellers, power will be generated.

The arrival of variable speed power conversion technology has opened up new and exciting sources of energy for development.

Aluminum Bronze - Method of Manufacture
Can Effect Performance

David F. Medley [1]

Abstract

Aluminum bronze alloys have displayed excellent wear and corrosion resistant characteristics in a variety of applications. Some grades of this family of copper based alloys can be produced with mechanical properties, resistance to galling, wear, erosion, cavitation and corrosion equal to, and in some instances, superior to those of stainless steel.

The in service performance of components can be enhanced by close control of chemistry and cooling rates. Conversely, failures can occur if necessary controls are not implemented in the manufacturing process. This particularly applies to welded fabrications where control of interpass temperatures is critical. Post weld temper anneal is generally desirable to preserve not only strength and ductility but also corrosion resistance. Although chemical composition is different from stainless steel, the two alloys are in close proximity in the galvanic series, thus coupling these alloys does not normally pose a corrosion problem.

Aluminum bronze alloys are perhaps the most versatile of copper alloy families. Their high strength and corrosion resistance combined with good anti-galling and cavitation resistance has resulted in these alloys being used for a variety of engineering applications. Most copper alloys are generally regarded as being low strength, difficult to weld and not readily available in most forms desired by the engineer. Aluminum bronze alloys are readily available in wrought and cast forms and have displayed a history of success, being used in valves, pipelines, condensers, pressure vessels, pumps and hydraulic turbines.

[1] SCOT FORGE CO., 8001 Winn Road, Spring Grove, IL 60081, (708) 587-1000

As shown in Table 1, it can be seen that aluminum bronze alloys rival the mechanical properties of stainless steels. However, when comparing materials, it is important to take into account all characteristics. Mechanical properties obtained from Standards Handbooks are representative only of test bars prepared under ideal conditions, whilst actual properties of a component can, in many instances, bear no resemblance to "test bar" figures. It is clearly illustrated in Fig. 1, that mechanical properties of some alloys are dramatically influenced by increase in section thickness, whilst others are less affected (Dralle 1977). It is interesting to note exactly what causes properties in cast tin bronzes to deteriorate so rapidly as their thickness increases, whilst aluminum bronze remains virtually unaffected.

TABLE 1

COMPARATIVE MECHANICAL PROPERTIES (TYPICAL)

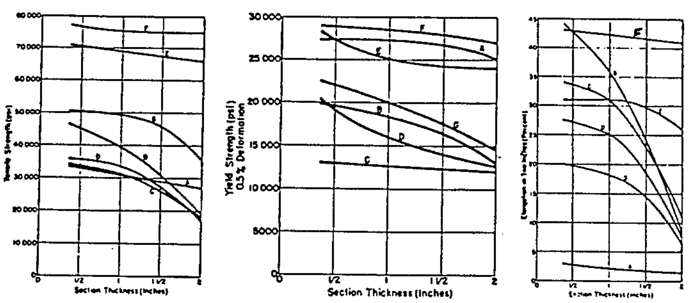
Alloy	316SS	C95200	C95400	C95500	C63000
Tensile ksi	80	80	85	100	100
Yield ksi	40	27	35	45	60
Elongation %	40	35	18	12	15
BHN	179	125	175	195	190

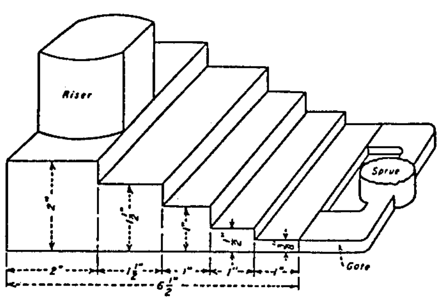
Two phase diagrams are illustrated in Fig. 2, one represents a range of aluminum bronze alloys, the other tin bronzes. Both diagrams are simplified, as most systems have more than two elements. However, the only area of comparison in this instance is the freezing range, that is the time/temperature difference through which all alloys pass in transforming from liquid to solid. It is very apparent that for aluminum bronze the difference is only some 12 to 22°F, but for tin bronze, it is very wide - some 200 to 300° F. Because of this, tin bronze castings are subject to dendritic growth with resulting micro porosity. As the alloy cools slowly, long needle-like crystals form. They are called dendrites and grow, not unlike deciduous tree branches. The "branches" which grow first, cut off the flow of molten metal, leaving small areas of porosity between the dendrites. The resulting structure often causes tin bronzes to weep under pressure test and deform under relatively light loading.

Aluminum bronze, which cools rapidly, has a dense equiaxed structure which is capable of retaining pressure and resisting loads up to its full range of published mechanical strength. Its cast properties are relatively unaffected by increased section thickness.

Whilst aluminum bronze alloys possess excellent mechanical properties and characteristics, these can be adversely affected by failure to impose manufacturing controls both in the melting and subsequent fabrication processes. Aluminum bronzes can generally be divided into 2 groups, those containing nickel and nickel free alloys. Both usually contain iron to lend strength to the alloys.

FIG. 1





Sketch shows general design features of step bar casting.

Composition and Minimum Mechanical Properties of Test Bars

	Aluminum Alloy	Tin Bronze 88-8-4	Red Brass	Tin Bronze 85-5-5-5	Manganese Bronze	Aluminum Bronze	Silicon Bronze
ALLOY DESIGNATION	A[1]	B	C	D	E	F	G
SIMILAR ASTM SPECIFICATION	B26–48T SC51A	B143–49T 2B	— —	B145–49 4A	B147–49 8A	B148–49 9A	B198–49 12A
NOMINAL CHEMICAL COMP., %	Cu — 1.25 Si — 5.00 Mg — 0.50 Al — 93.25	Cu — 88.0 Sn — 8.0 Zn — 4.0	Cu — 87.0 Sn — 3.0 Pb — 2.5 Zn — 7.0	Cu — 85.0 Sn — 5.0 Pb — 5.0 Zn — 5.0	Cu — 59 00 Mn — 0.80 Fe — 1.25 Al — 0 80 Zn — Bal.	Cu — 88.5 Al — 9.0 Fe — 2.5	Cu — 93.7 Si — 4.5 Fe — 1.8
MIN MECHANICAL PROPERTIES[2] Tensile Strength, Psi Yield Strength, Psi Elong in 2 In., %	32,000 20,000 2.0	35,000 18,000 20	27,000 12,000 20	30,000 14,000 20	60,000 22,000 25	65,000 25,000 20	45,000 20,000 13

[1] *Alloy "A" was tested in the solution treated and aged condition.*
[2] *½ in. dia sand cast test bars.*

Those containing less than 8% aluminum are single phase and are usually available in wrought form only. Other, more commonly used alloys nominally contain between 9 and 11% aluminum, are martensitic in nature, and possess all the characteristics encountered in metastable alloys. The characteristics of aluminum bronze alloys are affected not only by chemical composition, but by thermal control. Figure 3 illustrates the influence of aluminum content and cooling rate on the corrosion resistance (Macken & Smith 1961). Corrosion resistance and mechanical properties are affected by the type of crystal structure forming within the alloy. Figure 2 shows that if aluminum bronze alloys containing more than 9.3% aluminum are allowed to cool slowly from temperatures above 1050°F, the tough body centered cubic beta crystal transforms to the brittle eutectoid gamma 2. This not only affects the corrosion resistance of these alloys but results in a dramatic loss of ductility.

To some extent the addition of iron and particularly the addition of nickel will retard, but not eliminate, the formation of gamma 2. In order to eliminate the presence of gamma 2 in these alloys, both nickel free and those containing nickel, it is necessary to implement a temper anneal. This comprises of a rapid air cool to prevent the transformation which occurs at a temperature range between 1050°F and 800°F.

FIG. 2

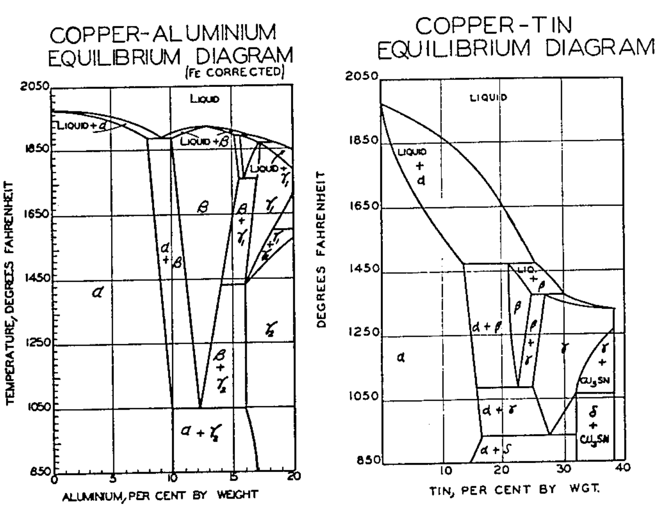
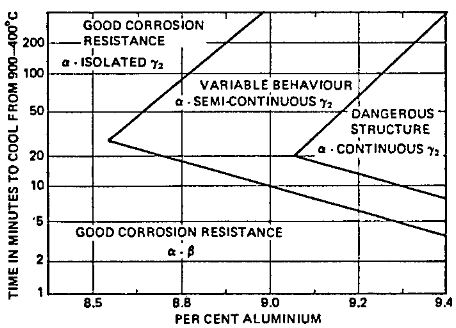


FIG. 3

Influence of aluminium content and cooling rate on the corrosion resistance of binary copper-aluminium alloys.



Temper anneal has benefits other than the elimination of harmful eutectoid. In nickel aluminum bronzes particularly, a refining of the grain structure from a coarser rosette type structure to a denser lamellar structure results. This is particularly beneficial for the subsequent ductility of the alloy and significantly improves weldability, in both nickel containing and nickel free aluminum bronzes.

As stated previously, aluminum bronze alloys which are not single phase, are martensitic, possessing a thermo plastic nature. Expressed in simpler terms, they have "memories". This martensitic nature increases as aluminum content increases. If these alloys are cold formed, they can subsequently attempt to return to their original shape. This "memory" can be triggered by input of mechanical or thermal energy, such as may occur in machining, welding or even by vibration encountered in service. Where cold forming is unavoidable, a full solution anneal may be advisable.

A most important control to be implemented is that of chemical composition. Most commercial compositions allow a total aluminum content spread of up to 2%. This is far too wide to ensure properly controlled microstructures even with the implementation of subsequent thermal treatment (Severson 1975). In order to achieve optimum strength and ductility, and ensure the end user consistent quality , alloys should have aluminum contents controlled to within +/- 0.2% of the desired amount. It should be appreciated that the desired amount of Aluminum can be varied by the skilled manufacturer to produce specific characteristics for specific applications. However, in the case of pumps and turbines some of the following guidelines may be worth consideration.

Copper Alloy Nos.	C61900 (Wrought)	C95200 (Cast)
Composition - %		
	Min	Max
Aluminum	8.5	9.5
Iron	2.5	4.0
Copper	Balance	Balance

An alloy possessing fairly good resistance to wear and galling. Below 9% aluminum the alloy tends to become coarse grained and its formability and weldability begin to deteriorate.

Recommended desired aluminum content is 9.1%. This allows manufacture of the alloy without necessity for temper anneal and allows a good degree of cold forming to be carried out without subsequent full solution anneal. The alloy is readily weldable at this composition and subsequent post weld thermal treatment is not necessary. Nominal iron content should be 3%.

Copper Alloy Nos.	C62400 (Wrought)	C95400 (Cast)
Composition - %		
	Min	Max
Aluminum	10.0	11.5
Iron	3.0	5.0
Copper	Balance	Balance

An alloy possessing good resistance to wear, galling and cavitation. Above 11% Aluminum, the alloy lacks ductility and weldability deteriorates. Recommended desired Aluminum content is 10.5%. Temper anneal is necessary after casting, hot working, or welding. Cold forming is not recommended. Nominal iron contend should be 4%.

Copper Alloy Nos.	C63000 (Wrought)	C95500 (Cast)
Composition - %		
	Min	Max
Aluminum	10.0	11.5
Nickel	3.0	5.5
Iron	3.0	5.0
Copper	Balance	Balance

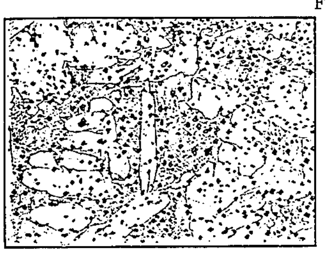
An alloy possessing good resistance to wear and galling, with excellent corrosion resistance and cavitation resistance superior to alloys C95200 and C95400. This alloy is in close proximity to high alloy austenitic stainless steels in the galvanic series. When considering a material couple between aluminum bronze and stainless steel, C95500 would be a logical selection.

Recommended desired aluminum content is 10.5%. However, equally important is the ratio of nickel to aluminum and content of nickel to iron (Weill Couly 1973). Therefore the following should be imposed:

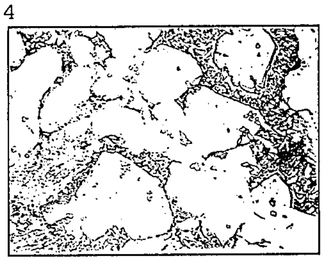
$$\text{Nickel} > \text{Iron}$$
$$\text{Aluminum} \leq 8.2 + \frac{Ni}{2}$$

Temper anneal is recommended after casting or hot working but post weld heat treatment is not always required if interpass temperatures are controlled and the alloy is not for service under severe corrosive conditions. Cold Forming is not recommended. Where cold forming cannot be avoided, alloy C95800 should be used, but caution should be exercised. Nominal iron content should be 4%.

FIG. 4



MAGNIFICATION 500X
Microstructure: alpha, beta, and a uniform distribution of (intermetallic compound).



MAGNIFICATION 500X
Microstructure: alpha, beta, eutectoid, and a non-uniform distribution of intermetallic compound.

An important purpose of adding iron to aluminum bronze is to form an extremely hard intermetallic compound, Fe_3Al (Severson 1975). When pressure is applied to a component, these fine particles lend strength and ductility to the alloy. The distribution of the intermetallic is achieved by pre-alloying a master alloy, generally 60/20/20 Cu/Fe/Al. This is added in molten form to the copper aluminum melt. Recent practices at many foundries has been to add molten iron to the copper aluminum bath. The result is that larger particles of free iron are formed, rather than a true intermetallic. Where this practice is followed, it is not unusual to see "rust spots" on the surface of the alloy. Obviously small particles homogeneously dispersed are better than large particles grouped together (Fig. 4), in contributing to the alloys strength and ductility.

In conclusion, it can be fairly stated that skilled manufacturers can produce controlled alloys to the benefit of end users and many manufacturers will provide technical support and advise on suitable alloys for particular engineering applications.

REFERENCES

(1)	Carl W. Dralle	"Inherent Corrosion Resistance and Improved Strength Have Broadened the Engineering Applications of Copper Base Alloys" - 15th Annual Liberty Bell Corrosion Course No. 2 - 1977.
(2)	P.J. Macken and A.A. Smith	"The Aluminum Bronzes, Properties and Production Processes". - CDA Publication No. 31 - 1961.
(3)	R.J. Severson	"The Role of Copper in Steel Production" Publication No 48/V - Centre Belge d'Information du Cuivre - 1975.
(4)	P. Weill - Couly	"Aluminum Bronzes and Their Uses" - 1973 CDA Conference on Plant Fabrication.

BALANCING TECHNIQUES - RUNNER IMPELLER REPLACEMENTS

K. David Bahn *
Jeffery L. Kepler **

Abstract

The objective of balancing is to reduce shaft vibration to a practical minimum. Reducing shaft vibrations generally reduces system bearing loads and increases the overall service life of the hydro turbine. Large hydraulic units are subjected to many kinds of vibrations. Vibrations and unbalance can be caused by mechanical, electrical, or hydraulic influences. At this time, the subject of mechanical runner impeller unbalance will be our primary concern.

The cause of mechanical unbalance is a shift of the center of mass away from the axis of rotation of the system. The center of this shift can be measured by two methods. These methods are static and dynamic analysis methods. There is great controversy on which method produces better results and is economically superior. Each method has its own advantages and disadvantages and each can easily be perceived as inferior. It is the purpose of this paper to look deeper into the balancing process. We will look into the methods and procedures that accompany each type of unbalance and attempt to describe the advantages and disadvantages of each method. With this information, it will be possible to determine which process produces the best technical and economic results.

* Manager, Facilities and Mfg. Technology, Voith Hydro, Inc., P.O. Box 712, York, PA 17405

** Associate Manufacturing Engineer, Voith Hydro, Inc., P.O. Box 712 York, PA 17405

Balance Equipment and Procedures

The static and dynamic balancing techniques each require specific equipment and procedures. Since static balancing is performed with the component in a stationary position, the equipment required is less complex than for dynamic balancing, which requires the component to be rotating. As a result, the overall cost of static balancing is about 40% of the cost for dynamic balancing. The following sections describes the equipment and procedure for static balancing and dynamic balancing.

Static Balancing

Static balancing is performed by using load cells to locate the center of gravity and jacks for handling the component. Three load cells are used because three points define a perfect plane. Jacks sufficient to handle the weight of the component are used to support the piece prior to balancing. The load cells are positioned in a circular pattern by an adapter plate that orients the circle of load cells to the component axis of rotation. During the first portion of the procedure, the jacks are supporting the component. At this time, the load cells are checked with the readout to assure proper zero adjustment and calibration. The load cells are also leveled to assure accurate readings. When the component is ready to be balanced, the jacks are lowered and the weight is transferred to the load cells. The cells are checked again to assure a level surface and the readings are obtained. The component is raised and rotated 180 degrees and the balancing procedure is repeated. With this rotational method, the errors are eliminated because the results are acquired by taking half of the algebraic difference of the two sets of readings. Any consistent errors that are present in the system will cancel out in the mathematical equation. This procedure assures the accuracy in the balancing technique.

Dynamic Balancing

Dynamic balancing is performed on a machine that rotates the component about the impeller's designed rotational axis. This machine consists of a precision universal coupling with an integral spindle that transfers the rotationally induced forces to a load cell or an electronic pickup. The component is first rotated at 1 rpm to establish a static balance condition. The component is next rotated at 15 rpm to provide significant input by rotationally generated forces. These rotational or dynamic forces are measured through the same load cell pickup and monitored electronically. The readings are synchronized at 90 degree positions and vector analysis is used to calculate the required corrections which, upon completion, is then applied to the component for re-balance.

To begin evaluating the two methods, we must look closer at the cause of a mechanical unbalance. When the center of mass of the system is offset from the axis of rotation, it produces a centrifugal force that increases as the unit's speed increases. Because of this, the centrifugal forces need to be minimized. International Standards Organization (REF ISO 1940-1973(E)) has issued a set of balance standards that show the level of balancing recommended for various types of rotating machinery.

Static and dynamic balancing are the same except for the fact that dynamic balancing uses centrifugal forces to cause the unbalance to be split between two planes. During static unbalance analysis the angular position of the unbalance is known, but the height of the unbalance is unknown. In the best scenario, the unbalance will be located at the crown (Plane-A)(Figure 1). This would increase the bearing load because the centrifugal force is further from the bearing, thus creating a larger moment. The location of this unbalance is the problem with this method of analysis. With a static analysis, it is possible that both the crown area and the band area would have a very large unbalance. If these unbalances would happen to be on opposite sides of the rotational axis, it would not be detected using the static method. This hidden unbalance can be detected if a dynamic analysis is used. This method rotates the impeller at a rotational speed which causes the unbalances to produce a centrifugal force. This method allows the impeller to be analyzed as a two plane impeller instead of the single plane that is used by the static method. The unbalance can then be distinguished between the two planes. Because of this, it is possible to identify and correct the unbalance that remains invisible to those who only use static balancing.

A simple example will be used to better compare the two methods. Since the real reason to balance the impeller is to reduce the bearing loads caused by impeller vibrations, the effectiveness of each balance method will be measured by determining the overall operational impeller bearing load. This will give a very realistic comparison between the two methods of balancing. The example used is a standard pump/turbine with an outside diameter of 150 inches and a total impeller weight of 35,000 lbs. As a reference, this impeller would be used for a net head of approximately 250 ft. The case specifications are listed in Figure 1. To begin, a typical static and dynamic analysis is completed. For each method, the impeller is balanced to the ISO standard G16 (REF ISO 1940-1973(E)). The bearing loads are computed at operating speeds to determine the effectiveness of the methods.

Outside Diameter	=	150 inches
Radius of Plane-A	=	50 inches
Radius of Plane-B	=	40 inches
Y1	=	60 inches
Y2	=	50 inches
Y3	=	300 inches
Operating Speed	=	300 rpm
Impeller Weight	=	35000 lbs
Balance Standard	=	700 in-lbs

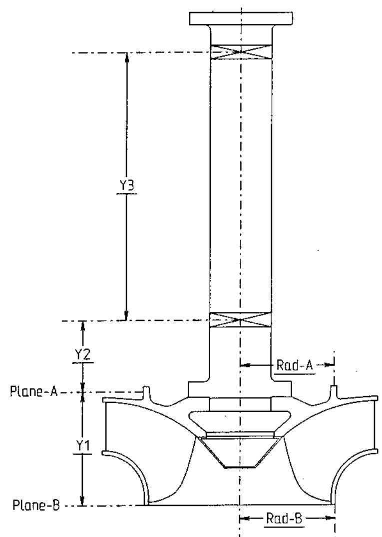


Figure 1
Case Specifications

Example Static Analysis

For the static analysis, some simple assumptions are made. Initially, the unbalance is assumed to be at the worst case location. This means that the unbalance is located at the band (Figure 2a). This is the worst case because its centrifugal force will produce the largest moment when the system is operating (Figure 2b). This unbalance is also assumed to be the level recommended by ISO G16, which is 700 in-lbs. With some simple dynamic calculations the operational bearing loads were calculated. The results show that the worst case static balance will produce a bearing load of 29,323 lbs. When the best case is assumed, the unbalance is located at the crown (Figure 3a). If the unbalance was located at the crown, the resultant centrifugal force would cause a smaller moment on the shaft (Figure 3b). This would cause the bearing load to be reduced to 25,032 lbs. The location of this unbalance is the problem with this method. As previously mentioned, with a static analysis it is possible that both the crown and the band would have a very large unbalance (Figure 4a). If these unbalances are in opposite locations it will not be detected using the static method. To show the possible effect of this type of unbalance, an

unbalance of 700 in-lbs is now placed at both the crown and the band to simulate the addition of a large hidden unbalance to the worst case condition. With a static analysis, these will not be detected, but it will cause the operational bearing load to increase from 29,323 lbs to 33,614 lbs. This proves that this hidden unbalance may cause an impeller to operate with higher bearing loads.

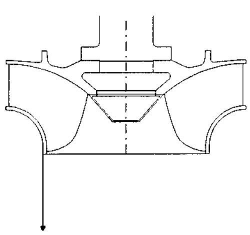


Figure 2a
Worst Case Static

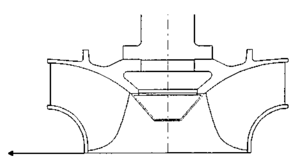


Figure 2b
Centrifugal Force

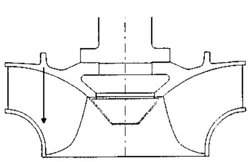


Figure 3a
Best Case Static

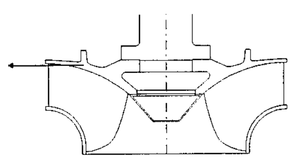


Figure 3b
Centrifugal Force

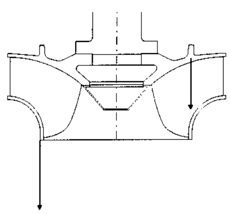


Figure 4a
Hidden Unbalances

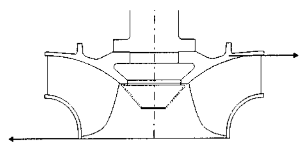


Figure 4b
Centrifugal Forces

Example Dynamic Analysis

For the dynamic example, the operational bearing load is also computed similar to the static analysis described above. The worst case is when the entire dynamic unbalance occurs at the band. This is the same situation as the worst case static that was discussed earlier (Figure 2a). And, if the unbalance was located at the crown, it would be the same as the best case discussed for the static analysis (Figure 3a). The big difference between the best and worst dynamic case is not the magnitude or the height of the unbalance, but the angular position. To simulate this, the 700 in-lbs unbalance is assumed to be split between the crown and the band, which gives each plane a 350 in-lb unbalance. Both unbalances are also assumed to be at the same angle (Figure 5a). This produces two centrifugal forces acting in the same direction (Figure 5b). The resultant bearing load is 27,177 lbs. If they are moved opposite to each other (Figure 6a), the produced centrifugal forces will be opposing each other (Figure 6b). Because of this, the bearing load drops to a level of 2,145 lbs. At this level, the impeller is balanced far better than what is really needed for a hydro turbine with an operational speed of 300 rpm. If the dynamic unbalances are located on opposite sides of the vertical axis it should be acceptable to balance the impeller to a standard of ISO G100 (0.12 lbs-in/lb.). This will produce a bearing load of 25,000 lbs, which is still far better than the level achieved by the static method. This acceptable dynamic level will vary depending on the height and the speed of the runner because as these increase, the moment due to the dynamic unbalance will increase. For example, if the height of the impeller was 150 inches (12.5 ft.) instead of 60 inches, it would only be necessary to dynamically balance the runner to ISO G40 (0.05 lb.-in./lb.). Also, if the speed is less than 300 rpm the dynamic unbalance forces are negligible if care has been taken in the manufacturing process to minimize these dynamic forces. This will work as a safe level because most hydro turbine impellers are much smaller then 12.5 feet in height.

The recommended method of balancing uses a combination of the two methods. The static balance method should be used initially to achieve

the level that is recommended by the ISO. Then the dynamic method should be used to verify that the manufacturing process has not left any hidden dynamic unbalances. This hidden dynamic unbalance needs to be reduced to ISO G40.

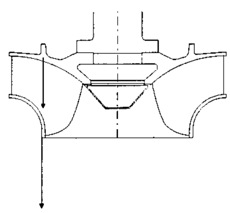


Figure 5a
Same Angle

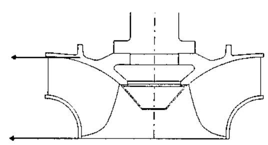


Figure 5b
Centrifugal Forces

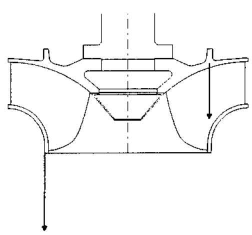


Figure 6a
Opposite Angle

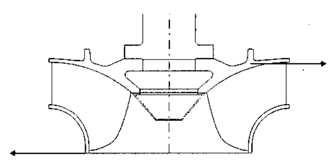


Figure 6b
Centrifugal Forces

Conclusion

The important thing to realize is that static balancing produces a similar result to dynamic balancing but it costs less. The only benefit of dynamic balancing is that it permits the detection of a possible hidden unbalance that is left undetected by a static analysis. To best utilize resources, the dynamic method should only be used to detect and remove large hidden values. For the remainder of the balancing operation, it should be sufficient to use the static analysis method. Regardless of which procedure is used, it must be realized that one of the most important steps in achieving a good balance is to consider the final balance during the entire manufacturing process. This will verify that all of the components have uniformity. This greatly reduces the hidden dynamic unbalance thus eliminating the need for dynamic balancing.

RECOVERING TECHNIQUES AND POSSIBLE USE OF THE GEOMETRY OF OLD RUNNERS

André Coutu[1]
Stuart T. Coulson[2]

Abstract

This paper describes the method used at GE Canada to recover the geometry of old hydraulic turbine runners. The measuring equipment will be discussed in addition to its use in the particular case of runners. Even if the paper focus is on the design of a replacement runner, other ways of processing the data by modern computer tools will also be presented with specific examples.

Introduction

For the last ten years or so, manufacturers of hydraulic turbines have invested heavily in computerized tools which are now widely used throughout the industry. These tools ensure a very high level of accuracy in all the steps of the development of a turbine, from design, to analysis and to machining. The end result is new machines with maximized output, manufactured at tolerances tighter than ever before (Ramchandani and Hsieh, 1992).

A problem is how to apply all these new technologically advanced tools, developed for new designs, to existing machines. In fact, very often the geometry of the old turbine components are unknown, particularly in the case of the runner. In this paper, we will describe how these unknown geometries can be recovered and fed into the modern computerized tools

[1] Senior Engineer, GE Canada, Hydro Business
[2]Hydraulic Turbine Design Engineer, GE Canada, Hydro Business, 795 First Avenue, Lachine(Québec), Canada H8S 2S8

available. Different possibilities in the utilization of the data will be shown with a focus on the design of a replacement runner.

Measuring a runner

To measure a runner, we use an optical inspection system as shown in figure 1. Composed of two theodolites linked to a computer, this portable system offers the possibility to measure the three coordinates of points with an accuracy better than 25 microns according to the manufacturer.

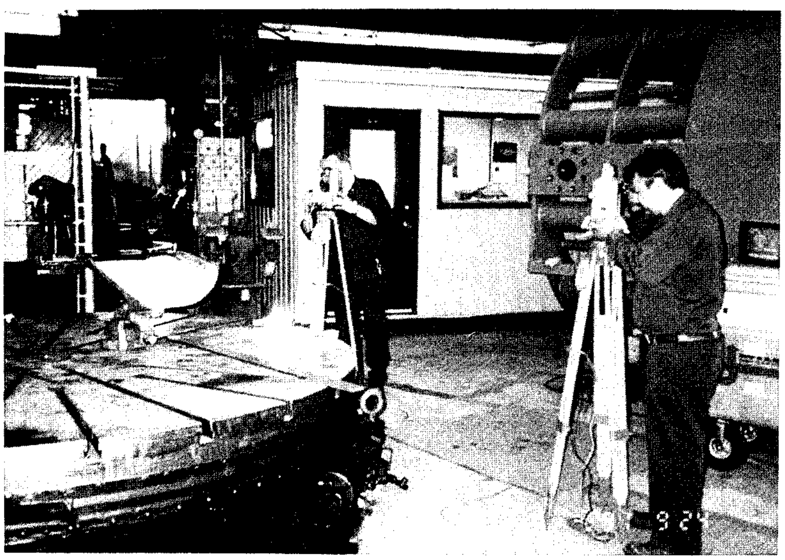


Figure 1. The Optical Inspection System

Points are first marked on the blade according to a predefined pattern (Coutu and Murry, 1991 and Coutu 1992) as shown on the blade of figure 2. The points are measured with the theodolites and the data is output, under any coordinate system and format, to a computer file compatible with our computerized hydraulic design system.

Easily usable with a runner sitting on the floor of the power house as shown on figure 2, this measuring method can also be applied to an installed runner. On figure 3, you can see the two theodolites positioned in the hole, ready to measure a marked blade. The unit was stopped only for the measurements.

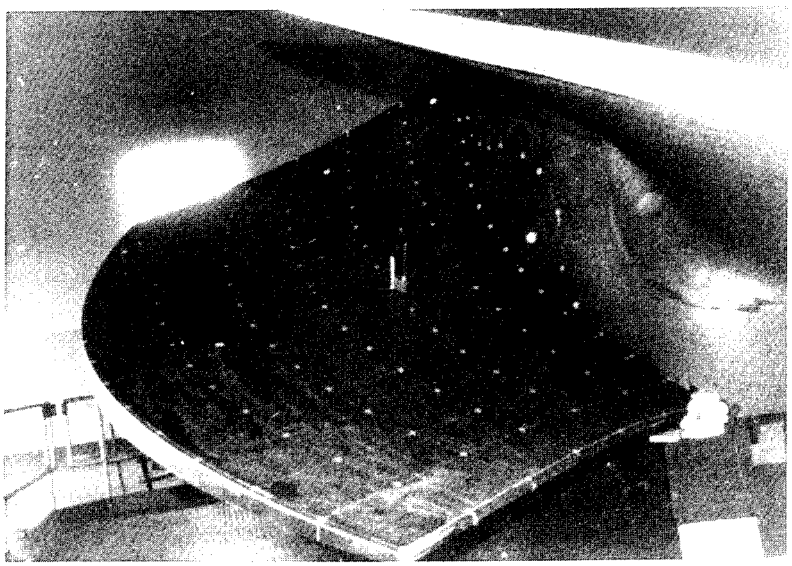


Figure 2. Blade marked for Measurements

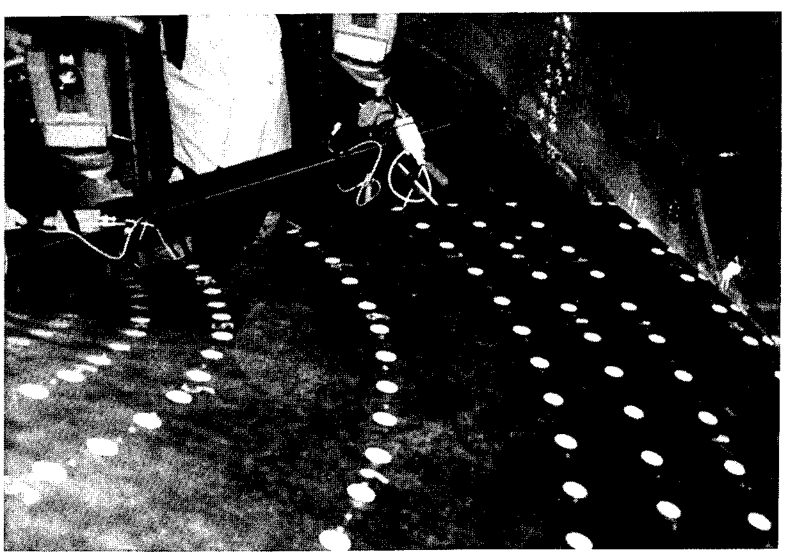


Figure 3. Installation to measure a Blade in the Hole

Using the same method, it is also possible to measure a smaller machine, as shown for a Francis blade on figure 1. It is the blade of a 750 mm throat diameter runner, and the next section explains the case in detail.

Design of a replacement runner

Runner replacement work typically provides significant challenges for both hydraulic and mechanical designers. Very often the drawings are either incomplete, illegible or even inaccurate due to undocumented modifications over the years. The following case presented a unique challenge since even the operating condition of the old unit was unknown. The turbine had not run for years and the owner of the powerhouse had no idea how much power it produced since the markings on the generator label had long since worn off. Nevertheless, he wanted the runner replaced with a modern design which would produce the same power as the original, to avoid burning the generator.

The unit was composed of two Francis runners, on a single horizontal shaft, which would discharge facing each other into a common tee branch. The casing was simply a cylindrical tank. The rest of the components were far from "ideal", involving a number of links, rods, and pins in the flow passage. The magnitude of the losses due to these surrounding components cannot be neglected and is very important to the design of new runners. Without homologous model tests, there would be no way to quantify these losses with any precision. For small hydro, model testing is of course out of the question, since the cost of these tests is often greater than the value of the contract.

The solution was to analyze the geometry of the existing runner. A blade (or in this case a bucket) was cut out of the runner and sent to GE Canada for measurement. The original condition of the blade is displayed in figure 4. The first task was to rebuild the surfaces to their original shape with a plastic material and mark the points to be measured (Figure 5). The blade was then measured (Figure 1) and the data fed into our computerized design system called *Biblades* (Do, 1989). This system is the fully interactive computerized tool used by our hydraulic engineers for the design, analysis and modification of runner blades. The computer model of the old runner is shown in figure 6. This model was then analyzed with GE Canada's flow analysis programs (Holmes and McNabb, 1982). The analytical results were used to determine the optimum operating condition of the existing unit as well as its acceptable cavitation limits, and these conditions were then used as the basis for the computer design of the new blade displayed in figure 7.

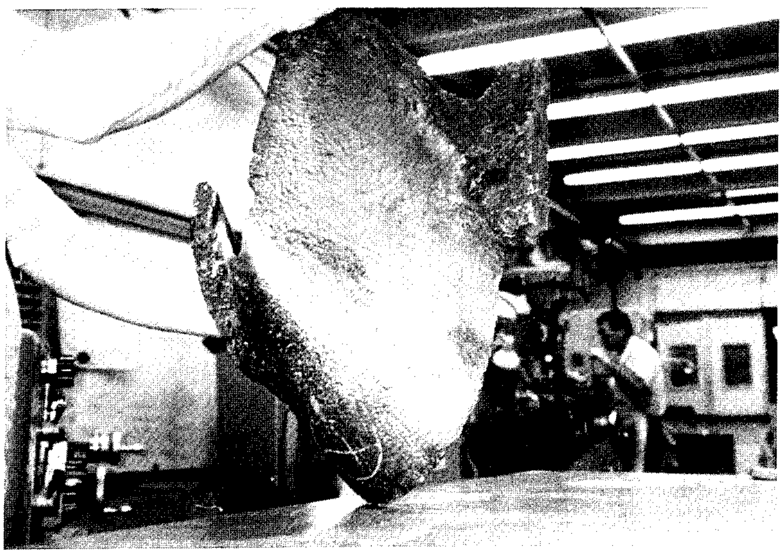


Figure 4. Old Blade as Received

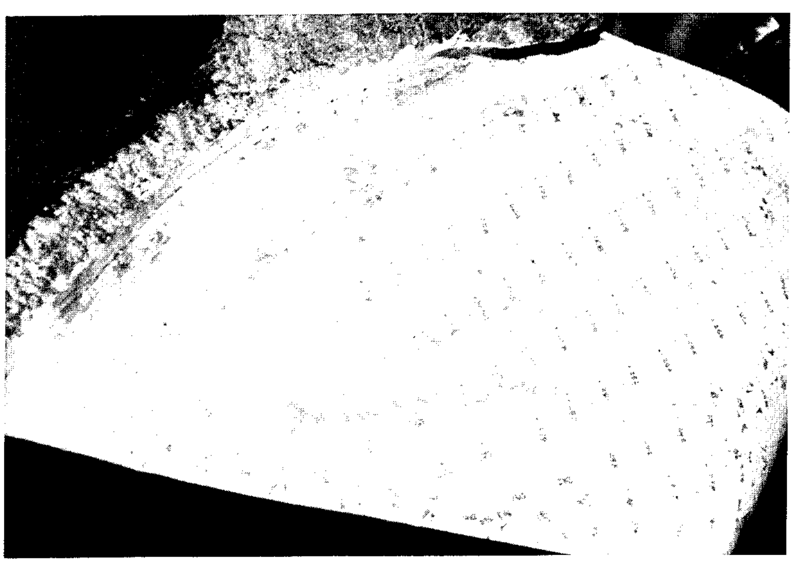


Figure 5. Old Blade with Surface Rebuilt

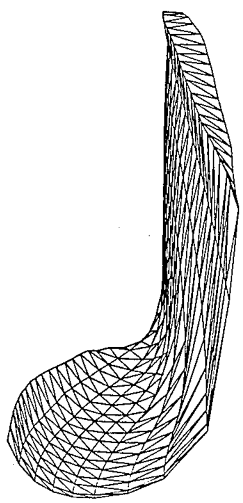


Figure 6. Computer Model of the Old Blade

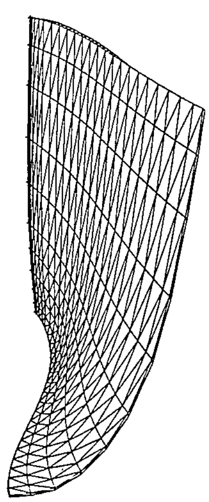


Figure 7. Computer Model Of the New Design

How to use the geometry of an old runner

Why would anyone ask to recover the geometry of an old runner? The answer may vary from one person to the next. The previous section showed that the development of a replacement runner is a reason. Another one is the production of a model runner (Coutu and Murry, 1991). The justification in this case is to compare the performance of a replacement runner with that of the existing machine by doing fully homologous model tests with comparison of the two runners. This is regarded as the most accurate method to measure performance gains by manufacturers (Demers and Désy, 1990 and 1992), by hydroelectric utilities (Markovich and Yap, 1991) , and by independent laboratories (Henry and Mombelli, 1991). The accuracy of the machining is guaranteed by our five axis machines (Murry, 1989). Figure 8 shows the model produced with the data measured from the runner of figure 2. The blades were machined individually and assembled to the hub.

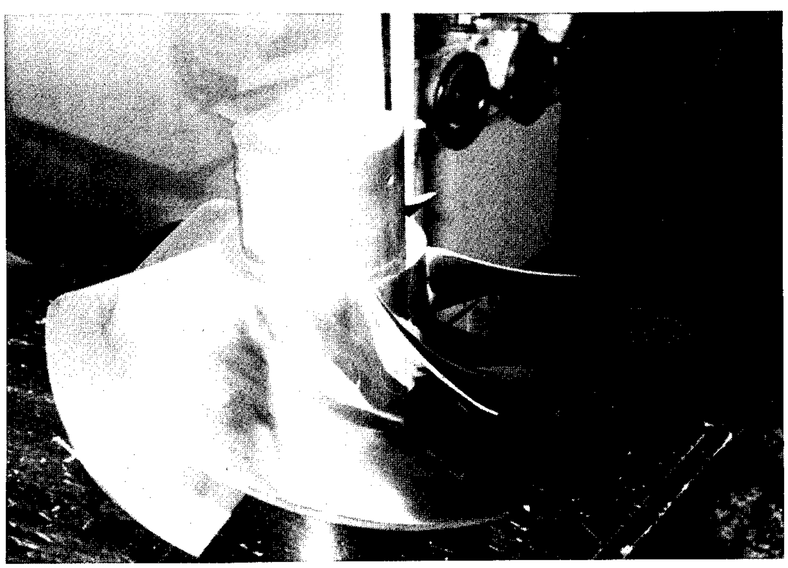


Figure 8. Model of a Measured Runner

Another reason to recover the geometry may be to study the mechanical behavior of the runner. Stress analysis can be performed on the geometries of runners (Coutu, 1990, Nuon and Paré, 1992) as shown on figures 9 and 10. Another customer may want to explore the possibilities of modifying an existing runner to increase its performance. By using our flow analysis programs , our specialists can discover the hidden possibilities of an old machine.

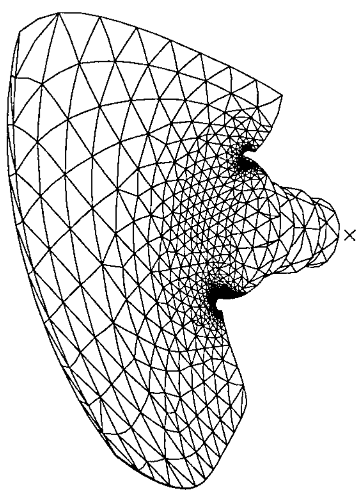


Figure 9. Finite Element Model of a Kaplan Blade

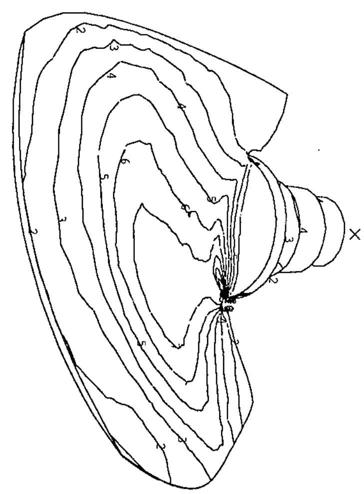


Figure 10. Stress Results for the Kaplan Blade Model

Finally, a good reason may be simply to reproduce the design under the format used before the computer age: the corebox drawing, as shown on figure 11 for the runner of figure 2. But the corebox drawing produced from a recovered geometry is more than simply the original design. All the distortions and modifications performed on the runner over the years are included in this drawing: it is the exact existing blade as it operates in the field.

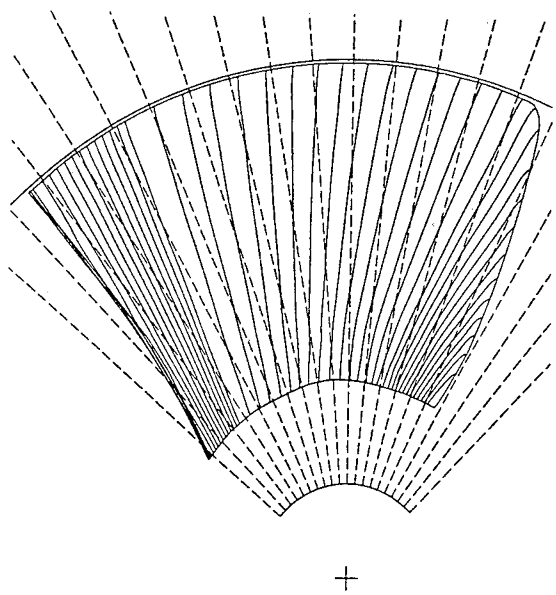


Figure 11. Corebox Drawing of a Measured Blade

Conclusion

We have presented a method to recover the geometry of runners. We have seen that the operation is possible with all kinds of geometry sizes and shapes. Some of the applications which can be performed with the measured data have also been shown. Although it may appear relatively simple, the overall process requires many integrated computer tools to be successful. All these tools are now available at GE Canada.

References

COUTU, A. - Finite Element Modelling of Periodic Parts for Stress Analysis, 15th Symposium of the IAHR, Belgrade, Yugoslavia, 1990.

COUTU, A., MURRY, N.S. - *Blade Model Generation for Existing Runners of Unknown Shape,* Uprating and Refurbishing Hydro Powerplants III, International Water Power & Dam Construction, Innsbruck, Austria, 1991.

COUTU, A. - *Experience in Recovering the Geometry of Turbine Runners,* 16th Symposium of the IAHR, Saõ Paulo, Brazil, 1992.

DEMERS, A., DÉSY, N. - *Experience with Hydraulic performance for Major Hydro Modernization Projects,* 15th Symposium of the IAHR, Belgrade, Yugoslavia, 1990.

DEMERS, A. DÉSY, N. - *Experience with Bersimis II and Chelsea Reporting Projects,* Sixth ASME International Symposium on Hydro Power Fluid Machinery, Anaheim, California, USA, 1992.

DO, H. - *Experience with the Application of Fully 3D Potential Flow Analysis to Runner Design,* IRCHMB-IAHR International Symposium on Large Hydraulic Machinery & Associated Equipments, Beijing, China, 1989.

HENRY, P., MOMBELLI, H.-P. - *The Interest of Comparative Model Tests,* Uprating and Refurbishing Hydro Powerplants III, International Water Power & Dam Construction, Innsbruck, Austria, 1991.

HOLMES, G., MCNABB, J.Y. - *Application of Three Dimensional Finite Element Potential Flow Analysis to Hydraulic Turbines,* IAHR International Symposium on Refined Modeling Flows, Paris, France, 1982.

MARKOVICH, M.S., YAP, R.W. - *Strategy for Achieving Optimum Performance in Turbine Runner Replacement/Upgrading,* Uprating and Refurbishing Hydro Powerplants III, International Water Power & Dam Construction, Innsbruck, Austria, 1991.

MURRY, N.S. - *Some new Approaches to the Solid Modeling and Machining of Hydraulic Turbine Runners,* IRCHMB-IAHR International Symposium on Large Hydraulic Machinery & Associated Equipments, Beijing, China, 1989.

NUON, N., PARÉ, D. - *Recent Developments in Finite Element Modeling and Stress Analysis for Hydraulic Turbine Components,* 16th Symposium of the IAHR, Saõ Paulo, Brazil, 1992.

RAMCHANDANI, H., HSIEH, R. - *Influence of Manufacturing Tolerances on Hydraulic Turbine performance,*Sixth ASME International Symposium on Hydro Power Fluid Machinery, Anaheim, California, USA, 1992.

PUTTING YOUR REPUTATION ON THE SLANT

by

Russell L. Katherman[1]

ABSTRACT

Restoration of the largest inclined-axis turbine units in the United States is a continual challenge and an opportunity which few in the hydropower industry will ever receive. The uniqueness of the six 26.7 kW, adjustable pitch Kaplan units, located at the Harry S. Truman Powerplant on the Osage River near Warsaw, Missouri, lies in the fact they are the largest of their kind to combine pumped-storage and inclined-axis technology.

Several units have experienced failures in the areas of high blade trunnion bushing wear, blade seal sleeves, blade control linkage and stub shaft coupling bolt studs. These failures are directly attributable to poor component fabrication and inadequate design for the situations in which they are being operated.

Business concerns with the necessary experience for rehabilitation of large hydropower equipment are not in great abundance in the United States. This results in a learning process for both the Corps of Engineers and involved parties from the private sector in order to handle the types of transportation, machining of existing parts and fabrication of new ones required by the renovation at Truman Powerplant.

The types of failures resulting from the coalition of poor component fabrication/engineering and an inclined-axis position presents difficulties for which optimization and innovative approaches are required to achieve an ultimate, quality rehabilitation.

This paper will describe the failures which have occurred, innovative directions taken in management of the project, engineering ideas, contracting methodology, new materials, and machining processes used to define overall rehabilitation of the six units as an eventual success. The information presented in this study will be of interest to all organizations that design Kaplan-type hydroturbines and/or currently operate Baldwin-Lima-Hamilton manufactured propeller units.

---

[1] Mechanical Engineer, Design Branch, U.S. Army Corps of Engineers, Kansas City District, 835 Federal Building, 601 East 12th Street, Kansas City, Missouri, 64106

INTRODUCTION TO THE POWERPLANT

The Flood Control Act of 1954 authorized construction of the project under the name of Kaysinger Bluff Dam & Reservoir. In March of 1967, both House and Senate Appropriations Committees were apprised that the most feasible powerplant was a 160 MW, pumpback, slant-axis design. The name of the project was changed in May, 1970, to Harry S. Truman Dam & Reservoir, and power generation began at the end of 1979.

The six pump-turbines were designed and manufactured by the Baldwin-Lima-Hamilton Corporation (BLH) to have adjustable turbine blades from zero (closed) to 28 degrees pitch opening in conjunction with the wicket gates. The units themselves are inclined at an angle of 24 degrees from horizontal and supported by a lower (stub) shaft and a two-section main shaft with an unsupported first span of approximately 39 feet. Oil lubricated bearings, cooled by water from the reservoir, support the shafts at the pier nose structure, main shaft packing box, generator shaft coupling, and at the end of the unit. The single thrust bearing located at the extreme downstream portion is double acting with a rated design load capacity of 1.3 million pounds in both directions.

The six motor/generators were designed and manufactured by the General Electric Corporation and are rated at 13.8 kV, 3 phase, 60 Hz for 100 rpm of the turbine for both motor and generator functions. Each motor/generator is rated at 36,500hp in generation and 42,200hp at 49 feet of total dynamic head with a flow rate of 4,500 cfs.

During construction and subsequent operation of the powerplant, many concerns were expressed by the state of Missouri, those benefiting from power generation and concerned local citizens regarding the procedures and methods of operation by which the project was intended to function. The problems which developed were:

a) Unacceptable fish mortality which resulted during testing of the pumpback feature.
b) Concerns for downstream impacts of power generation.
c) Power consumers might experience higher energy rates with the project operating at less-than-full capacity.

The latter concern of functioning at less-than-full capacity was originally limited by the selection of an operating plan which dictated that power generation would be by four (4) units from the design power pool, without pumpback, that the Corps of Engineers would work toward full generation in phases and that detrimental downstream effects would continually be monitored. Pumpback is in reserve until technology can provide a reliable source of removing fish from the tailrace area.

With all of the obstructions regarding operation the of Truman project, the last circumstance desired was failure of the hydro-turbines, the very items which must demonstrate the reasons for expenditure of taxpayer funds. Unfortunately, this occurred, and the ensuing burden on the Corps' excellent reputation was momentarily eclipsed. This trend is being reversed with much effort and devotion to correcting the situation.

The turbine failures discussed and their associated repairs will be presented in the following format: events which occurred, analysis of the components from which the events resulted, contractor efforts toward rehabilitation, Corps assumption of repair obligations and the resulting benefits of this experience.

PREVIOUS PROBLEMS AND UNIT FAILURES

Technology, as it was 25 years ago when Congress approved the hydropower function, seems narrow in its application of turbine knowledge to a slant-axis design. It would appear that many of the components used are designed for horizontal Kaplan units and were simply turned to the desired angle upon installation (refer to Figure 1). This has proven detrimental to some of the systems involved and hazardous to those who must maintain them.

On December 1, 1985, Units #6 and #3 experienced vibration warnings which necessitated removing them from service for performance of inspections to determine the cause(s) of the alarms. The Government then contracted disassembly and analysis functions from Newport News Industrial Corporation (NNI) for renovation of these units.

Subsequent investigation showed that both units experienced fatigue failures in the eye bolt portions of their linkage control systems. The related blade, on the five-bladed units, lost angular control and pitched radically, damaging some of the internal components. This loss of blade function necessitated immediate analysis of the water tightness of the powerplant as it related to the draft tube liner and the possibility of rupturing it by the loss of blade restraints on a rotating turbine.

Analysis showed the eye bolts were of arguably poor design for the conditions in which they were installed. It was apparent that the highest stresses occurred during a blade opening stroke transition from 0° to 14°. As the units were disassembled it was discovered that other conditions of operation would be susceptible to failure in the future if their present conditions were not corrected.

The failure of the blade #2 eye bolt for Unit #6 was determined to be caused by high-cyclic, low-stress fatigue in the radius transition between the shoulder and shank of the bolt on the side with an anti-rotational dowel pin. This transition may be typical of the BLH design for this style unit.

Metallurgical examination was performed on the eye bolt to determine its chemical composition, hardness, tensile and impact strengths, microstructure and fractography of its failure surfaces. Chemical percentages showed the material to be approximately AISI 1045 steel, and 11 of 12 yield tests gave a value of 11.7 ksi less than the required 55 ksi. The impact tests gave an average value of 6.2 ft/lbs at 0°F and 32°F with the latter appearing to be the nil ductility transition temperature. The remaining four eye bolts from Unit #6 were examined by wet fluorescent magnetic particle, ultrasonic and liquid dye penetrant inspection procedures with the mag-particle method giving the only indication of a defect. Each of the four eye bolts showed small cracks in the transition area described.

Unit #3 was inspected at the same juncture by use of a baroscope and determined to have suffered the same failure as Unit #6 on blade #4. After determination of the cause of the Unit #6 failure, the blades on Unit #3 were blocked in position and the turbine was returned to operation until it could be renovated. In March of 1988, Unit #2 also lost control of a blade. The cause is assumed to be the same as the two previous units but has not been conclusively identified to date. At this point, all units were provided with blocks to restrict blade movement. The blades are set in the most efficient operating mode, but the units are now subject to very rough starts and stops.

The blade journal bushings were discovered to be highly loaded due

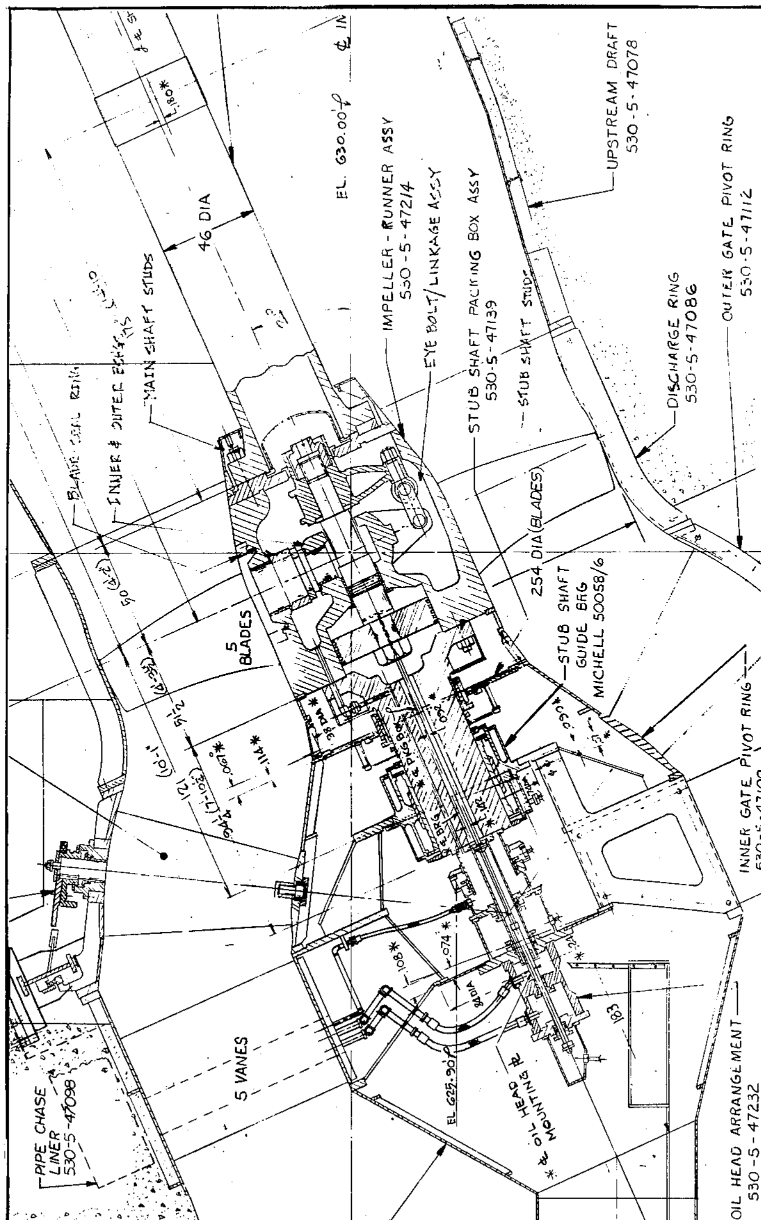


Fig. 1 - Section through Pump/Turbine Assy.

to the short moment arm from the inner to outer journal surfaces with distinctive wear patterns of a trapezoidal shape. The bushings were also found to be burnished and bronze transfer was occurring to the blades journal surfaces. Chemical tests showed much lower lead content (actual 4.99-5.57% versus the designated levels of 8.0-11.0%) which resulted in a higher hardness than was requested in the original design. The detrimental results of this situation were easily detectable.

During this time, the 10 main shaft (downstream) and 15 stub shaft (upstream) coupling bolt studs were analyzed for the appropriate pre-loading stress condition which would ensure an infinite fatigue life for these bolts. Previously, Unit #4 experienced an emergency shutdown due to extreme stub shaft run-out. This was due to failure of seven of the fifteen stub shaft bolts. The remaining eight were non-destructively tested, of which seven were identified to possess cracks. New bolts were subsequently installed in Units #2 and #4. No failures of main shaft coupling bolts have occurred.

In January, 1992, Unit #5 was noticed to be loosing inordinate amounts of runner oil. The unit was unwatered and an investigation showed the oil to be coming from the blade packing area. Upon removing the blade packing, it was discovered the stainless steel packing seal ring mounted on the blade had fractured. The ring was subjected to mechanical and chemical analysis which showed it to be manufactured according the original requirements of type 410 stainless steel. Visual inspection of the ring demonstrated the extent to which stresses on the blade bushings for blocked units are being subjected. The packing is in a chevron stacked pattern and made from cotton impregnated nitrile rubber material with a texture pattern which was impressed upon the ring's surface, although the ring hardness was in the range of Rockwell_c 32.6 to 37.5. Reservoir water quality analysis displayed high chloride levels which, when coupled with cyclic loading on the ring due to the blade's blocked position, created a stress corrosion cracking situation. It is anticipated, as bushing wear continues in the blocked condition, that unless replaced, the other units will suffer similar failures.

Blocked blades on Kaplan turbines appears to be creating other areas of concern. In March, 1992, Unit #3 blades were noticed to have cracks at the inside position of the two blocking sets which had been installed at each blade. Two blocks have been installed on either side of every blade on each unit and are mounted at approximate midway points from the centerline of the blade journals. While propeller units are known to crack and lose portions of their blades, it appears that blade blocking may be inducing additional cyclic fatigue stresses in the most susceptible locations.

Unit #6 had repairs performed to it and was returned to service in January, 1990, and disassembly of Unit #3 proceeded. After 1313 hours of operation, Unit #6 was again removed from service due to excessive oil leakage from the oil head location. Investigation and disassembly revealed five problem areas on this recently repaired unit: blade/discharge ring contact; dry lubricant contamination; bronze bushing particle contamination; water in the runner hub; unsatisfactory bushing oil grooves. At this juncture, January, 1991, the Corps decided to perform the repair work with its own forces, including all project management, construction, contracting, parts procurements, quality control and engineering functions.

ENGINEERING AND MACHINING

The loss of blade control by eye bolt failure was corrected by procurement of new eye bolts for all units made of AISI 4340 material forgings (145 ksi tensile, 125 ksi yield, 16% elongation, Charpy V-notch of 15 ft/lbs at 0°F) with the eye/shank transition radius increased from 1/8" to 1/2", shot peening of the transition area and die rolling of the shank threads in lieu of machining. Each bolt, upon installation was pre-stressed with a hydraulically loaded nut. The hydraulic nut is composed of two sections with an internal rubber bladder which was pressurized to obtain eye bolt elongation for a pre-load stress condition of 12.1 ksi. The use of hydraulic nuts also eliminated the anti-rotational dowel pin which had added to the failure area's stress concentration. These changes will result in the extension of endurance life of the eye bolts to such a degree as to preclude future cyclic fatigue failures.

Given the results of the blade journal bushing analysis, new inner and outer bushing castings made of ASTM B271-84 copper alloy with a modification to the lead content were provided. This was eliminate the high hardness condition. An increase in the number of oil lubrication grooves from nine to 33 in the outer bushings and from six to 16 in the inner bushings were also provided. In an attempt to improve the contact area of the bushings and thereby the load carrying characteristics, the new outer bushings were machined with an additional 0.007" eccentricity offset from the inner bushing (the bores of the runner hub already contain a 0.007" eccentricity offset) giving a total of 0.014" upstream "tilt" to the blades. The bushings were also supplied with a coating of dry film lubricant prior to blade insertion upon reassembly of Unit #6.

When Unit #6 experienced the five problem areas and was disassembled a second time in the summer of 1990, inspection of the blade bushing surfaces showed very sharp edges to the oil grooves which acted as oil wipers rather than oil supply grooves. All the groove edges were supposedly broken at a certain chamfer prior to blade insertion for oil transfer. The bronze and dry lubricant contamination was determined to be from the wearing action of the blade journals on the bushings since there appeared to be no oil reaching these surfaces during the short 1313 hours of operation after the first reassembly. The Corps, after deciding to perform the work on its own, procured new castings for Unit #6 and redesigned the number (14 from 33 in the outer bushings and 9 from 16 in the inner bushings) and shape of the oil grooves. The surface profile was shaped in a very rounded form for a radii of 5/16" at the groove to loaded area transitions. This new contour is expected to provide adequate oil supply without compromising the load-carrying capacities of the bushings.

The stub shaft coupling bolt studs will also be replaced in all units with AISI 4340 steel. The new studs will be provided with die rolled threads and mechanical properties of 145 ksi tensile, 123 ksi yield, 16% elongation and a Charpy V-notch impact value of 25 ft/lbs at 0°F. Pre-load stresses will be induced by thermal expansion of the studs using heater rods placed into holes drilled approximately three-quarters the depth of the studs. Scheduled intermittent heating cycles coupled with alternating nut torquing will provide permanent elongation of each stud so as to attain a pre-loaded stress value of 42.0 ksi. This level of pre-load stress is designed to provide an infinite fatigue life for the specific operating conditions at the powerplant. The main shaft coupling studs, while not experiencing nor exhibiting signs of failure, were subjected to an increased pre-load stress. However, upon subsequent fatigue life analysis and optimization by the Corps, the bolt material was

changed to AISI 4130 steel with mechanical properties of 110 ksi tensile, 90 ksi yield, 16% elongation. Pre-load stresses will also be induced by thermal expansion so as to attain a value of 24.0 ksi. Since the main shaft studs are considerably larger, a smaller preload level was required to produce the same infinite fatigue life curve as the stub shaft studs. Each stud will also be coated with manganese-phosphate to remove the possibilities of galling with the internal threads of the runner hub during installation.

Given the water quality of Truman reservoir and the nature of the lake runoff area being mainly farmlands, a new material for the blade seal ring surface must possess excellent chemical resistance to fertilizers and chlorides in addition to elevated mechanical properties to remain intact for the life of the turbines. The analysis showed that an ASTM A705/A705M, type XM-25 solution annealed forging would provide all the necessary requirements. Each blade will be provided with these new rings in the future as they are rehabilitated. However, the stainless steel seal rings for Unit #6 have not been replaced due to the time constraints for the repair, blades having never been blocked thereby inducing no cyclic stresses on the existing rings, and application of a vinyl paint coating on surfaces subject to water contact.

To date the blocked blade condition remains in effect for the three of the four operational turbines (#1, #2 and #4) although it was intended to be a short-term solution for allowing the turbines to be returned to service by removing the stresses from inadequate internal components. After discovery of the cracks in Unit #3 blades, no other defects were found but several blades had blocks broken away from their attachment location on the external surface of the runner hub in the operating turbines and others had various cracks in their weldments. Upon finding these conditions the blocking design was altered to provide additional contact area for the blades and reinforcement for the blocks. Caution is recommended to be used in the exercise of this procedure as the final effects of extended use of these units in this hampered condition is not fully known. While the fiscal benefits of having usable turbines is obvious, the damage and necessary repairs to the blades, runner hub and/or other pieces of equipment/systems may be extensive in the long term.

Other engineering improvements and optimizations made since the Corps' decision for performance of all work by in-house forces were machining and polishing of the blade journal surfaces and the blade bushings, increasing lubrication and material properties for the internal linkage components, use of polymer washers to attenuate fretting corrosion in the linkages, installing tapered dowels to the runner hub/adapter plate connection, polishing of internal threads of the runner hub, and hydraulic nuts coated with Mn-P.

The blade journals and stainless steel seal rings for Unit #6 were machined on a vertical mill to ensure concentricity of the three surfaces. Each was then polished, along with the inner and outer bushings, to better than a 32 RMS microinch finish. It was determined that since the lubrication in the journal surfaces is of boundary type, this would improve the oil supply to the region. In order to develop the most accurate and highest quality blade to bushing wear pattern and loading condition, the disassembled #6 runner was transported to a machining facility and placed on a horizontal boring mill. The inside diameters of the inner and outer bushing locations were measured along with verification of the 0.007" eccentricity centerline offsets of each of the five bore pairs. Each bushing was first machined then installed in the runner hub and line bored on the horizontal mill to allow for an exact 0.010" to 0.014" diametral clearance of the bushing bores with

respect to the blade journal surfaces and also provide for uniform oil groove depth throughout the loaded zones. The previous method for determining final bushing inside diameters relied upon calculating the amount of compression the bushings would receive upon insertion into the runner hub.

The linkage system pins were upgraded to 120 ksi tensile, 80 ksi, 17% elongation and hardened to an R_c of 32. The link plates were also given a reduced surface at either end to allow for flow of oil to the bushings in the eye bolts and lever arms. The polymer washers are designed to withstand the vibratory forces acting at both ends of the linkage system, thereby, eliminating the fretting occurring during unit operation.

Removal of the adapter plate from the runner hub is required for access to the internal components. There are fifteen 6" dowels around its circumference which must be removed by drilling out the ends or destructive methods. Provision of tapered dowels and tapered dowel holes with threaded plugs will allow for easier removal and reuse of both items if future internal work is required.

In July, 1992, during attachment of Unit #6 to the main shaft, four of the coupling stud bolts became seized in the internal runner hub threads. Attempts were made to remove them by liquid nitrogen but they eventually had to be machined out. Inspection showed the threads had cold-welded. Thread measurements indicated the sharp ridges on the internal and external treads may have contributed to development of burrs and metal particles. After consulting with Ozark Powerplant in Arkansas it was discovered this situation had occurred before at that location during coupling stud replacements. Based upon their experience, an internal "hone" was developed by removing the threads from an old coupling stud of the same thread configuration and attaching it to a cylinder hone in place of the polishing stones provided. Lapping compound in various grits was used to polish the internal main shaft and stub shaft coupling holes of the runner hub. Trial fitting of new and existing coupling studs proved this method to be very beneficial to the overall thread condition and will be performed on every turbine when coupling studs are removed and installed.

Use of the Mn-P coating was very successful in disrupting the galling affects of the alloy steels involved in the installation of the main shaft and stub shaft coupling bolt studs. All hydraulic nuts, two for each piston shaft and one for each eye bolt on every turbine has also received the coating per DoD-P-16232F, Type M, Class 2. Thickness ranges were allowed to be between 0.0005" to 0.0010". This is expected to ease installation of the components and reduce a potentially expensive hazard to the associated parts during reassembly.

MANAGEMENT AND CONTRACTING

It is traditional for the Corps of Engineers to produce plans and specifications and conduct construction administration for performance of large hydropower and pumping plant installation and repairs, while leaving the actual construction efforts to private parties under Government supervision. However, from the Corps' perspective, this method was not the most desirable one available in this situation since two turbines were already disassembled, others required rehabilitation and serious delays had already occurred. Confidence in the Corps was suspect at best leaving our reputation in this area open to possible criticism.

While all projects of this magnitude require good management and

delicate coordination of every aspect to be successful, the endeavor of the Corps' performing all facets of the repair was regarded with some trepidation due to the events and delays encountered during the previous five years of contractual work.

The first and foremost requirements of success are committed management and excellent personnel who will perform every point of the renovation with proficiency. With a project of this type, the initial issue which must be resolved and communicated to anyone and everyone who will be even remotely involved is priority. Where does this effort fall within a myriad of others? What will be the rank of funding over other projects? Are there safety concerns, loss of income, cost benefits, customer relationship affects and a host of other items which were defined in order to give the renovation a chance of succeeding? The Corps' management, in this regard, was excellent and creative in the sense that this project was given very high priority and the appropriate latitude was given the responsible parties for scheduling and progression of the work as was deemed necessary for the most efficient and highest quality end product.

Coordination of the various functions, i.e. technical requirements and contracting/purchasing, must be performed by someone who can communicate well with each department. In differing from most projects, scheduling efforts were not drafted with the intent of forcing others to comply with them. Discussions of the necessary labors involved with each aspect and component of turbine renovation, legal requirements, organizational regulations, and engineering evaluations and designs were taken into account in order to generate the most accurate and realistic schedule possible.

Engineering elements took the lead in identifying the areas of the turbines requiring rehabilitation but relied heavily upon those whose responsibility it is to operate the facility. Each area was discussed and then defined by the project coordinator who directed the items to be designed and developed in specified order for the purpose of taking the plans and specifications to the contracting/purchasing department for placement of the requisitions in the private sector for bids. While this is not unusual, it took people who were not afraid of accepting responsibility in breaking the normal mode of operation, expedite all orders and search for the most efficient methods possible for attaining the necessary supplies.

Machining of Unit #6 six blades and runner bushings in place was performed by AC Equipment Services, Inc. (presently Siemens Power Corporation) with very satisfactory results and was completed ahead of schedule. The unusual aspects in machining of the remaining five sets of turbine blades, runner hubs, adapter plates and blade journal bushings is in the contracting method used. The contract was developed in such a manner as to allow tremendous flexibility for scheduling and ordering of specific work items as required. The contract, as awarded to Siemens Power Corporation, was a delivery order contract with Government exercised option years. In other terms, each individual component could be ordered separately as the turbines are disassembled. While each item is set in a specific pattern for each function to run concurrently on any particular turbine, it allows for schedule flexibility with regard to the Government's rehabilitation crew. Each turbine is not held to a specific time frame for which the components will become available for the specified work, and the contractor's limited duration for performance of the machine work does not begin until the Government requests such action. With four turbines awaiting disassembly and machine work as described and one presently being machined at writing of this paper, the entire turbine renovation project can be

mapped out within specific time frames but still allows for adjustments in the event of unanticipated problems. This lends itself to established costs for future fiscal years, preferable Government/contractor relationship and higher quality assurance. In addition, substantial savings in engineering and overhead costs are still being realized since replete plans and specifications for the repair of two disassembled units and renovation of four others was not required. Savings of at least one year in engineering time for production of a complete renovation package would have been required.

With the Corps performing its own quality assurance checks on the components procured and the machine work performed, problems have been reduced and non-conformance items are held to a minimum. While this approach requires many manhours of effort, productive assembly time has been increased. The project coordinator and contracting/purchasing functions must devote much attention to each order for the vendors to place the same high priority on the work as the requesting organization's management and project team. The advantages to doing this have been and are being realized in tremendous time and overhead savings as well as allowing for a speedier repair.

CONCLUDING REMARKS

The adage "putting your reputation on the line" was used in defining this project with the exception of "slant" being substituted at the end due to the orientation of Truman powerplant's turbines and because of the downward trend in their repair for the last several years. It can be said with certainty that the efforts and funds expended since the Kansas City District has taken on the project personally by in-house forces, the repair of these units, in order to provide our customers with a dependable source of energy, will be achieved in the very near future. With the existing dedication to every aspect of the renovation, optimistic and varied thinking toward the required functions, commitments to quality and timeliness, this project is moving forward at a rapid pace.

ACKNOWLEDGEMENTS

The subjects presented in this paper are the result of manner hours effort by many for the previous seven years. The author would like to thank the Management, Engineering, Operations and Contracting Divisions of the Kansas City District, Mechanical Power Section of the Omaha District, and the many others with whom he is involved on this project for their individual commitment, input and team effort in making the rehabilitation a continued success.

REFERENCES

(1) Stan Abramovitz, Abramovitz and Associates Inc.; Investigation of Runner Blade Bearing Failures, Harry S. Truman Powerplant, August 22, 1990.

(2) David B. Benson, Lead Engineer, Newport News Industrial Corporation; Engineering Analysis Report For Unit 6 Runner Failure, Revision A.

(3) Donald J. Juett and Hadley E. Wolfrum, Hydropower Branch, U.S. Army Corps of Engineers, Kansas City District; Synopsis of Harry S. Truman Turbine Repair, Waterpower Papers, pages 2019-2028, 1987.

(4) Robert A. Weber, Engineering and Materials Division, U.S. Army Construction Engineering Research Laboratory, Corps of Engineers; Failure Analysis of 410 Stainless Steel Seal Ring on Unit #5, Harry S. Truman Powerplant, March 19, 1992.

MODERNIZING A DOUBLE FRANCIS TURBINE - FLOW ANALYSIS AND HYDRAULIC MODEL TESTING ENHANCES TURBINE PERFORMANCE

Peter F. MAGAUER *

ABSTRACT

A case study showing the modernization of horizontal Francis turbines will be presented. It will be demonstrated that the performance of the units could be improved considerably after a process of intensive optimization based on hydraulic model testing.

The requirement to conserve the appearance of all important parts in the powerhouse represented a challenge to both hydraulic and design engineeres. The unusual model test arrangement involving an air test will be described in detail.

INTRODUCTION

In all industrialized countries, the demand for electric energy is increasing, while the public opinion is rather negative on constructing new powerplants, no matter which technology is applied. One of the few unquestioned sources of providing additional electricity is to modernize existing powerplants in order to gain the highest possible energy production from existing schemes.

Old hydroelectric powerstations often have waterways which are very different from today's practice. This makes it difficult for the equipment manufacturers to use already tested hydraulic models as a basis for bidding. Flow analysis based on advanced computer programmes is used with increasing success to support the work of experienced engineers.

For projects where performance is emphasized, however, physical model testing still guarantees the highest quality of the results, regarding both optimization of performance and confirmation of theoretical assumptions.

* Senior Engineer, Hydro Turbines and Plants,
VOEST-ALPINE M.C.E. , Linz, Austria

THE ANDELSBUCH POWERPLANT

In our case study, the rehabilitation of the Andelsbuch powerplant will be analyzed. First of all we will shortly describe the project in general.

The Andelsbuch powerplant is located in the very western part of Austria and has been built in the beginning of the century. Its task was to contribute to the electricity supply of the region including neighbouring countries. In particular it was intended to cover the increasing demand of the important textile industry as well as of private homes. The four double Francis turbines with horizontal axis have been supplied by Rusch-Ganahl, a local manufacturer, and started operation in 1908.

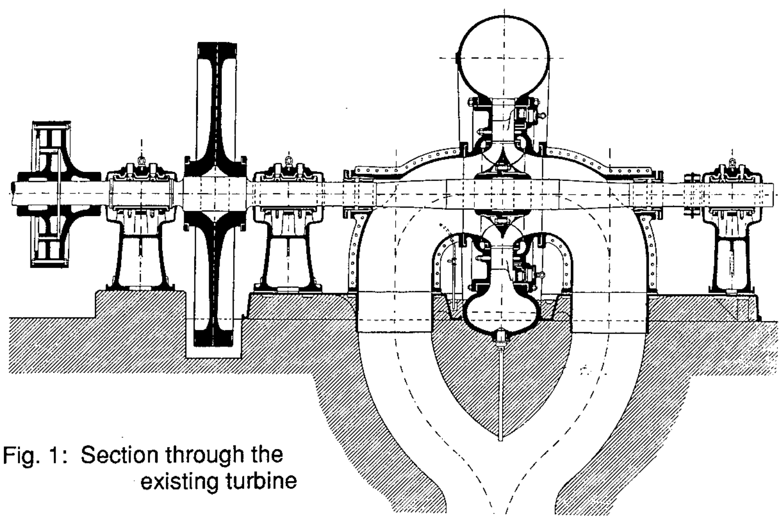


Fig. 1: Section through the existing turbine

Each double Francis turbine is fed through a common spiral case, the draft tube is designed with a bifurcation and a common draft tube elbow (Fig.1). A relatively large flywheel helps to improve the stability of the system. The most significant technical data are listed in table1.

TECHNICAL DATA :		
Rated output :	2182 kW	
Rated head :	59.9 m	(197 ft)
Synchronous speed :	500 rpm	
Runner diameter :	700 mm	(27.6 in)
Intake valve diameter :	1000 mm	(39.4 in)

Table 1 : Important technical data

TARGETS OF THE MODERNIZATION

Despite careful maintenance, the customer decided to rehabilitate the existing powerplant. The target was to increase output and efficiencies of the units without changes of the civil structures. The appearance of the whole scheme including dam, penstock and powerhouse had to remain unchanged.

The conservation of the historic character of both the outside and the inside of the powerhouse was an important requirement even for the electromechanical equipment. The historic appearance (Fig. 2) should not suffer from the modernization.

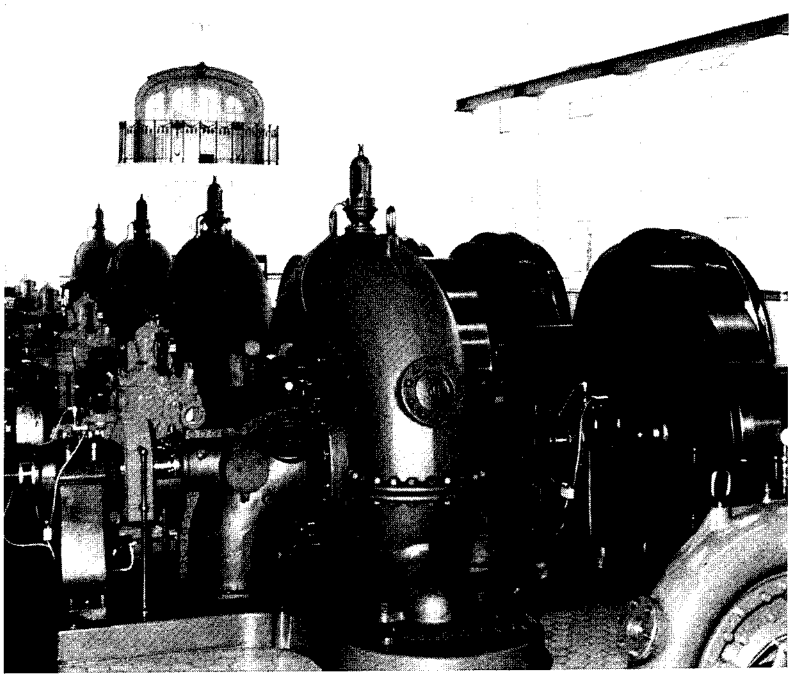


Fig.2 General view of the powerhouse floor

The specification asked for the installation of new turbines, whereas the existing draft tubes had to be integrated in the new configuration. The scope of supply comprised runner, shaft, distributor, spiral case, inlet valve, inlet pipe, seals and several minor parts. The governor will remain unchanged.

An increase of net head or discharge was not possible. The additional energy production had to come basically from increased efficiencies all over the range of operation. A weighted average efficiency has been evaluated acc. to the formula

$$\eta_{AV} = 0.05 * \eta_{50} + 0.08 * \eta_{70} + 0.10 * \eta_{80} + 0.45 * \eta_{90} + 0.32 * \eta_{100}$$

where η_{50} means turbine efficiency at 50% of maximal discharge, the other percentages have to be applied accordingly to the other values. Net head and tailwater level have been defined for each point. The target of the customer was to increase the average efficiency by at least 5 %.

The guaranteed efficiencies as quoted in the offer had to be verified in a model test which had to be homologous as far as possible. A price adjustment depending on the model test results has been agreed.

The requirement to test the turbine on a physical model represented an extraordinary challenge to the test engineers, as they had to find a sound method to test this special arrangement in a satisfying way.

CONCEPT FOR MODEL TESTING

A completely homologous model test was practically impossible due to the fact that the design differs very much from nowaday's practice. After investigating different possibilities, a combination of two test methods has been selected:

o A model test in air for draft tube elbow and draft tube bifurcation (Fig. 5)

o A model test in water for a single Francis turbine representing a hydraulic model for the double Francis prototype turbine (Fig. 4).

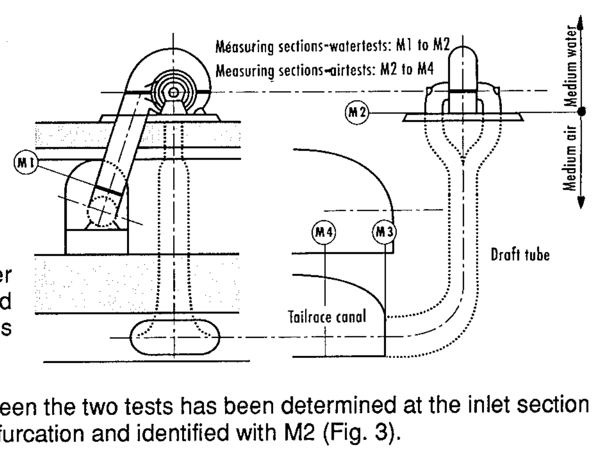
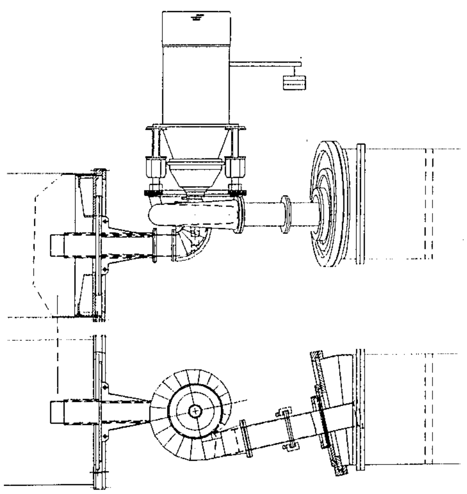


Fig. 3 : Water passage and reference sections

The interface between the two tests has been determined at the inlet section of the draft tube bifurcation and identified with M2 (Fig. 3).

For the correct combination of both tests it was important to define the test heads carefully as explained below. In order to allow a correct hydraulic model test, the air test had to be finished prior to the hydraulic model test.

DESCRIPTION OF THE MODEL TESTS



The tests have been performed in the hydraulic laboratory of Voest-Alpine M.C.E. in Linz/Austria.

This laboratory comprises five hydraulic test rigs with closed water circuits, including up-to date data acquisition and monitoring systems. For certain special problems there is an air test stand included in the test facilities.

Fig. 4: Hydraulic test installation

The target of the air test has been to determine the head loss coefficient for the downstream part of the water passage, starting from the interface section M2.

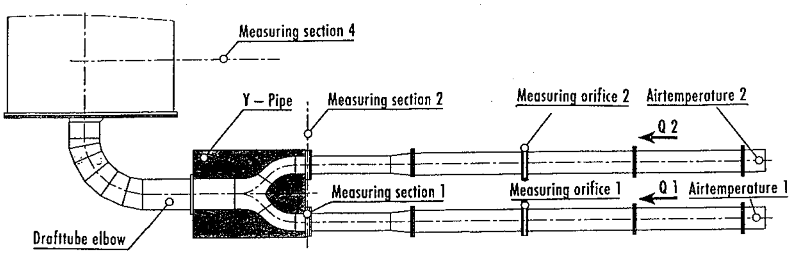


Fig. 5 Arrangement of the air model test

The actual air test series have been done on a model with scale 1:3.7. The 15 feet long upstream pipes have been applied to provide adequate measuring conditions with equalized flow.

The hydraulic model has been tested with a runner of 340 mm diameter (Fig.7), the prototype test head should be equal to the model test head. As the hydraulic test was related to the turbine without the draft tube section downstream of M2, the test head had to be reduced as follows:

$$H_{1\text{-}2} = H_{1\text{-}4} - H_{2\text{-}4} = H_N - \zeta * (Q/2)^2 / (A_2{}^2 * 2 * g)$$

with the following abbreviations:

$H_{1\text{-}2}$	test head turbine (water)	(m)
$H_{2\text{-}4}$	test head draft tube (air)	(m)
H_N	net head of prototype	(m)
ζ	head loss coefficient from air test	(-)
Q	discharge of prototype	(m^3/s)
A_2	sectional area at interface M2	(m^2)
g	gravity acceleration	(m/s^2)

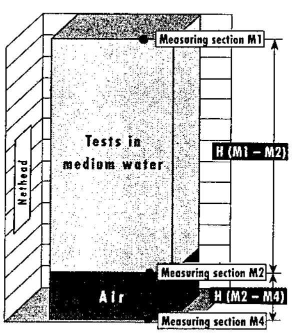


Fig. 6: Definition of test head

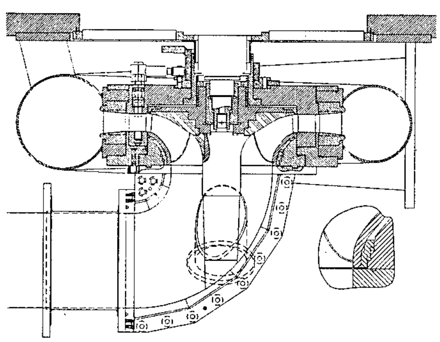


Fig. 7: Model turbine

All specific values for the hydraulic model test like unit speed n_{11}, unit discharge Q_{11}, and cavitation figure σ have to be calculated with above test head.

Efficiencies and cavitation behaviour have been tested internally, optimized and demonstrated in a witness test. Further, the runaway speed characteristics have been shown.

COMPUTATION OF TEST RESULTS

The results have been derived from the model tests using a clearly defined stepup procedure. A constant adder resulting from the arithmetic average of either formula Ackeret and Hutton had to be applied. The stepup value finally was 0.7% for all points.

For the computation of results, the following inputs have been used:

- o Test results of the hydraulic model test
- o Mechanical losses (bearing friction and ventilation losses)
- o A constant adder for the stepup to prototype conditions
- o Test results of the air test (draft tube section)

These influences lead to the following formula for the computation of eficiencies:

$$\eta_P = (\eta_M - \underbrace{P_L / P_P}_{\text{MECHANICAL LOSSES}} + \underbrace{0.7}_{\text{STEPUP}}) * \underbrace{(1 - \zeta * (Q_P/2)^2 / (A_2{}^2 * 2 * g * H_N))}_{\text{DEDUCT FOR AIR TEST SECTION (DRAFT TUBE PARTS)}}$$

with the following abbreviations :

η_P	prototype efficiency	(%)
η_M	model efficiency	(%)
P_P	prototype turbine output	(kW)
P_L	mechanical losses	(kW)
Q_P	prototype discharge	(m)

The friction losses between fluid and head cover appeared on the model only, but not on the double turbine of the prototype. This fact has been neglected for the computation and represents an additional safety margin.

MODIFICATIONS

In the bidding stage, it was necessary to derivate the guaranteed values from existing model tests, taking into consideration the estimated influences of the specific hydraulic conditions of the particular project.

It was a very encouraging event when the first measurement series according to the procedure described above showed results which have been very close to the guaranteed level.

Starting from the base model, several modifications have been carried out, aiming at an improvement of the weighted average efficiency. The most important steps have been carried out as follows (Fig. 9):

- o Modification of the stay ring:
  A joint consideration of flow analysis and stress computation led to a reduction of stay ring losses.

- o Modification of the wicket gates :
  Compared to the standard profile of the wicket gates, a new profile specially adapted to the project brought significant improvements.

- o Optimization of the runner :
  Supported by flow analysis and physical model testing, the runner has been optimized for the specific project conditions.The specific discharge has been increased, and the efficiency level could be raised while a satisfying cavitation behaviour has been maintained.

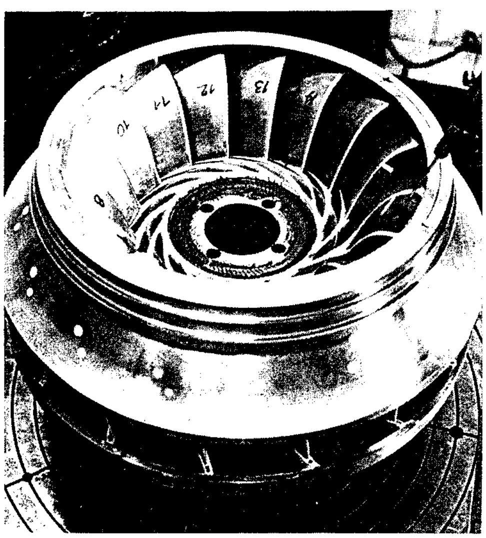


Fig. 8 : Model runner

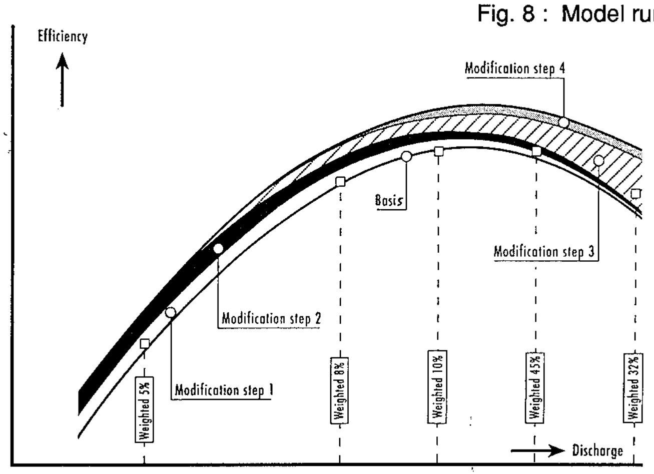


Fig. 9 : Steps of optimization

RESULTS OF MODEL TESTING

The combination of the modification steps as described above lead to a considerable improvement of the efficiencies. The guaranteed weighted average efficiency finally has been exceeded by more than 1.5%.

This improvement which has been an outcome of intensive activities on the hydraulic test laboratory has been honoured by the customer with a bonus payment of approximately 100 000 $.

As a further result of the intensive optimization work, progress has been made regarding the knowledge how to deal with specific modernization tasks. It was shown that model testing can provide an important contribution to a sound and optimized solution of a complex modernization task.

DESIGN FEATURES

The task for the design engineers then was to fulfill the following criteria for the layout of the components:

- Proper connection to the remaining old parts
- Conservation of the "old fashioned" exterior design of the units
- Easy handling during installation
- Reduced requirements for maintenance

For the spiral cases for example, cast steel material has been selected, mainly to cover the requirement for a "neat looking" interior appearance of the powerhouse.

The wicket gate mechanism has been delivered with self-lubricating bushings in order to avoid any danger of water pollution and to facilitate the maintenance of the units.

Special care has been taken to provide proper connections between old and new parts, in particular at the following interfaces:

o Spiral case -	foundation plate (existing, cast steel)
o Inlet valve -	penstock (existing, riveted plate steel)
o Turbine -	draft tube bifurcation
o Turbine shaft -	flywheel, bearings and generator
o Gate operating ring -	existing governor (mechanical type)

Design ideas and manufacturing methods have changed a lot since the old units have been built. The remaining old parts generally have been in in good condition, as they have been treated carefully and maintained intensively. Special attention had to be drawn on the integration of new and old components.

The turbines by now are in the design and manufacturing stage, the first unit is scheduled to re-start operation in early 1994.

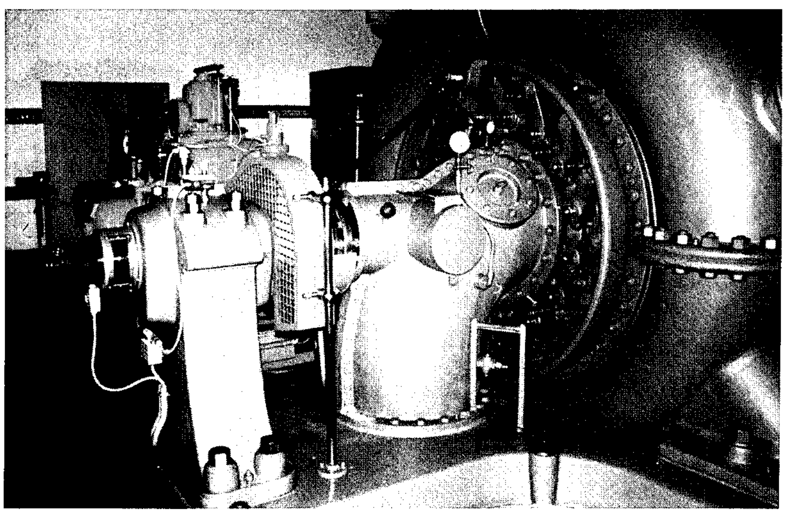


Fig. 10 Detail view of the existing turbine

CONCLUSION

The introduced case study showed the possibilities for a tailor-made, optimized modernization of a well kept old powerplant.

Computer aided flow analysis combined with extensive model testing contributed to a significant improvement of the performance.

We could demonstrate that even for unusual hydraulic configurations like the discussed double Francis units with common draft tube, a test method could be found which, by combining two partial tests, led to reliable results.

TURBINE PERFORMANCE TESTS AT THE ROBERT MOSES NIAGARA POWER PLANT

Robert J. Knowlton[1], Peter W. Ludewig[2]
Jack C. Howe[3] and John H. Phillips[4]

Abstract

As part of an upgrade of the Robert Moses Niagara Power Plant extensive laboratory model tests were conducted to optimize the design of new turbines, including tests of the original design. Field tests were conducted on original-design units and an upgraded unit to assure that the manufacturer of the new runner had achieved the contract guarantees and to examine the differences in prototype performance as compared to differences in model performance. Flows in the original and upgraded prototype units were measured using an acoustic method. The relative increase in turbine efficiency and the output level at which peak efficiency is achieved are similar to those expected from the laboratory model tests. In addition, results of overspeed tests conducted on the upgraded unit agree with computational predictions when model-derived parameters are used.

Introduction

The Robert Moses Niagara Power Plant (RMNPP) was completed in 1961 as part of the Niagara Power Project, and is located in Lewiston, New York. Studies conducted in 1987 indicated that the 13 units could be upgraded by replacing the turbine runners, modifying the generators, replacing the transformers, and making other modifications, so that higher capacities could be achieved (see reference 1 for additional details). Higher capacity operation would allow re-timing additional energy to peak demand periods,

---

1. Technical Advisor to the Vice President - Production, New York Power Authority, 123 Main Street, N.Y. 10601
2. Senior Engineer, Engineering - System Operations, New York Power Authority, 123 Main Street, N.Y. 10601
3. Senior Mechanical Engineer, Parsons-Main, Inc., Prudential Center, Boston, Mass. 02199
4. Superintendent of Power, Niagara Power Project, 5777 Lewiston Road, Lewiston, NY 14092

and provide additional firm capacity, when operated together with the Lewiston Pump Generating Plant, which also utilizes the RMNPP forebay.

Although increasing the capacity of the RMNPP units from about 175 to 200 megawatts generator output was a principal project objective during the runner procurement process, the high value of efficiency was recognized and incorporated into the model design competition (see reference 2). During the planning process, it was not certain that the best gate turbine efficiency could be improved, since the original turbines were considered to have excellent efficiencies and good cavitation performance. The extensive model design and testing competition described below was therefore undertaken.

Efficiency guarantees were provided by the competing manufacturers for both model and prototype performance, as required by the procurement specification. In addition, visual begin cavitation guarantees were required for the models, and runaway speed and hydraulic thrust limitations were established by the procurement specification for prototype machines.

Model Design Competition

A model design competition was conducted in 1989 at the Institut de Machines Hydrauliques et de Mecanique des Fluides (IMHEF) in Lausanne, Switzerland. For this competition, IMHEF fabricated a model including the intake, penstock, spiral case, stay ring, and draft tube.

Two competing contractors provided runner models which were tested by IMHEF staff. After the tests the most valuable runner was selected by the Power Authority and its consultant, Parsons-Main, in accordance with pre-established criteria (see reference 2). Voith Hydro was accordingly authorized to provide a prototype runner.

In addition to testing models of upgraded turbines, IMHEF also tested a model of the original design to provide an equivalent basis for comparing the performance of the upgraded turbine models. The results of testing the original design agreed reasonably well with the model tests conducted in 1958 prior to construction of the project.

The testing conducted by IMHEF on all three models included:

- Turbine efficiency over a net head range of 270 to 320 feet, exceeding the 280 to 310-foot net head range of operation. Each guarantee point was tested, and additional points tested to define fully the shapes of the efficiency curves.

- Wicket gate torque, including torque when one gate is not restrained.

- Cavitation, including inlet cavitation, visual observation of cavitation begin and development of sigma-break curves. Photographs were taken to provide a permanent record.

- Hydraulic Thrust.

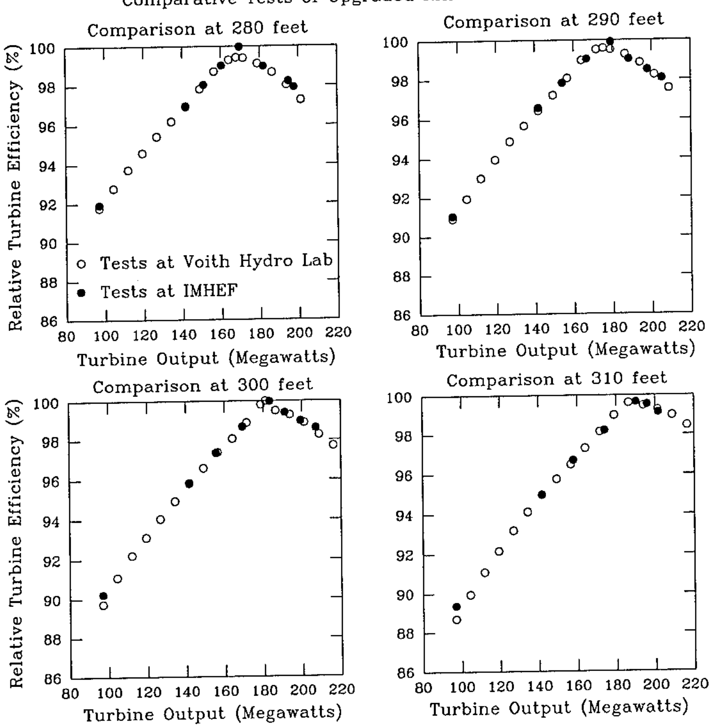
- Runaway speed at different sigma levels.

- Pressure pulsations at several locations in the penstock, spiral case and draft tube, and shaft torque fluctuations.

The contract documents also required that the turbine contractors conduct model tests in their own laboratories, including a witness

Figure 1

Comparative Tests of Upgraded RMNPP Turbine Models



Notes: Turbine efficiency is expected prototype efficiency based on a 2/3 Moody step-up: Ep=1-(1-Em)/3-((Dm/Dp)^0.2)(1-Em)/1.5

where Ep=Prototype Peak Efficiency Dp=Prototype Discharge Diameter

Em = Model Peak Efficiency Dm=Model Discharge Diameter

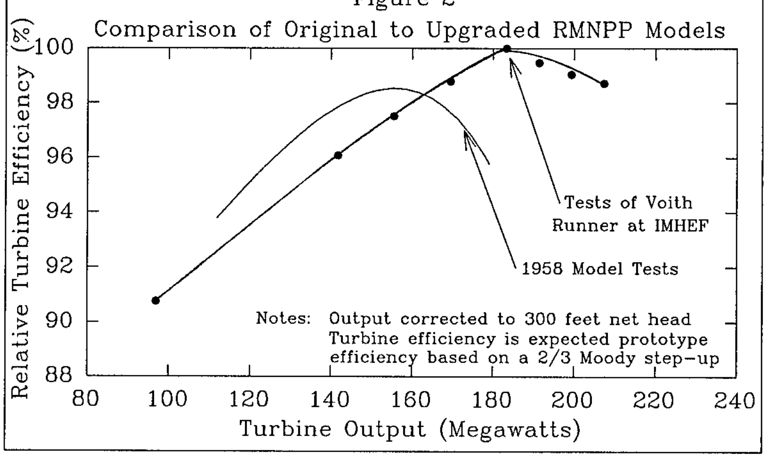
test, before delivering their best model runner to IMHEF for competitive testing.

Figure 1 provides a comparison of the results of model testing the runner design in Voith Hydro's hydraulic laboratory with the results obtained at IMHEF. It should be noted that the contractors were not required to include a model of the penstock in their model, but did fabricate their own homologous spiral case, wheel case and draft tube sections. Differences of a few tenths of a percent in efficiency are evident on Figure 1, and not all tests conducted in the manufacturers' laboratories agreed as well with the IMHEF tests as the efficiency curves plotted on Figure 1. This reinforces the idea that a design competition such as this is best accomplished by testing the competing models in the same independent laboratory.

Figure 2 provides a comparison of the expected prototype efficiency at 300 feet net head based on the original 1958 model tests and the IMHEF tests on the runner which became the basis for the prototypes. Generally, the model exceeded the guaranteed model efficiency levels. In both cases, a 2/3 Moody step-up is applied to efficiency (see Figure 1), with no shift in output.

Figure 2

Comparison of Original to Upgraded RMNPP Models



Physical Differences between Original and Upgraded Runners

To achieve the higher output levels as described in the Introduction, the upgraded stainless steel fabricated runner was designed to achieve its peak efficiency at a higher output, as seen on Figure 3. To accomplish this, and to provide the most efficient turbine possible, the upgraded runner has a longer band, larger outlet diameter and fewer blades than the original runner. Specifically, the upgraded runner has a discharge diameter of

208 inches, compared to the original 205 inches, and the band length increased from 40 inches to 65 inches.

Field Tests

Field efficiency tests were conducted in 1988 on RMNPP Units 3 and 4, and in 1992 on unit 4 after it was upgraded. Flow measurements were performed using permanently installed four-path two-plane acoustic flowmeters in each unit with a shared data acquisition system provided by Accusonic Division, ORE International, Inc. (see reference 3). The acoustic flow measurement method was specified as the official acceptance test method in the contract documents.

The acoustic flowmeters were installed since these could serve as part of on-line efficiency measurement systems in the future. Acoustic flow measurements are of particular interest since the measurements require less preparation than other methods applicable to this project, once the test equipment has been installed, and have demonstrated very high repeatability. Figure 3 shows the general layout of the plant and the measuring locations.

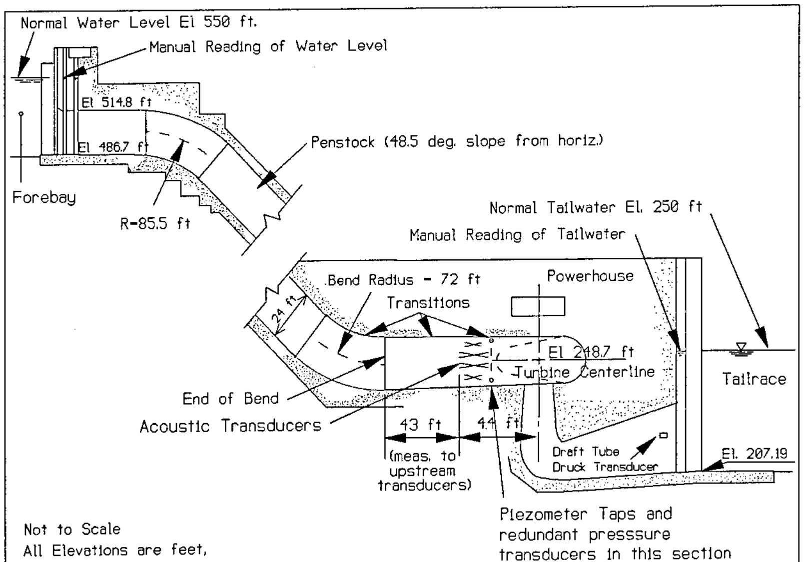
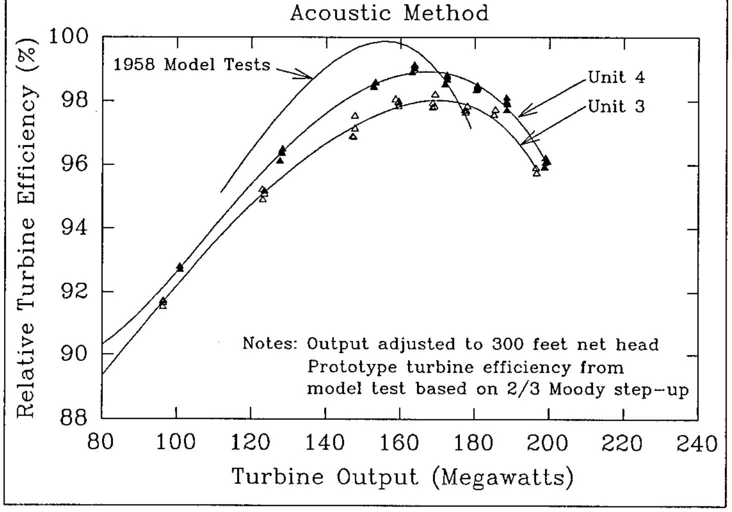


Figure 3
Robert Moses Niagara Power Plant
Hydraulic Passageways and Measuring Locations

Inlet heads were measured by connecting a Heise pressure gage to the piezometer manifold installed during original plant construction immediately upstream of the spiral case inlet. To provide redundancy, two Druck pressure transducers were installed in diagonally opposed positions at the piezometer tap section. Further cross-checks of the data were made possible by manually reading the unit headwater level and also recording the station headwater level during the tests. For the acceptance tests of the upgraded turbine, an average of the Heise and Druck pressure measurements, which generally agreed to within 0.1%, were used.

Tailwater Levels were measured by installing Druck transducers in each draft tube trifurcation, by manually reading the tailwater level during the tests, and by recording the station tailwater level during each test. Manual tailwater measurements were made using two 4-inch diameter PVC stilling wells with perforated bases, which were installed in the stop log gate slots in the outer trifurcations. For the acceptance tests of the upgraded turbine, the manual tailwater measurements were used.

Figure 4 – Comparison of Field Test Results
Acoustic Method



Generator output was measured using a 3-phase Yokogawa power meter and a 2-1/2 element Scientific Columbus power meter. A cross-check was provided by correlating unit output with power levels recorded in the switchyard approximately 3,500 feet away. Calorimetric tests were run before and after the prototype upgrade to determine generator efficiency on a consistent basis for use in turbine efficiency calculations.

Significant effort was made to assure that each measurement transducer was properly calibrated. For the pressure transducers, this meant calibrating over a range of headwater and tailwater levels to provide an accurate calibration relation for use in the data acquisition system.

During the tests, the forebay level was maintained as constant as possible. This is particularly relevant to the acoustic flow measurements, where measuring periods of 10 to 15 minutes are averaged to develop a single data point.

Results of Pre-Upgrade Efficiency Testing

Figure 4 provides a comparison of the efficiency results using the acoustic flowmeter in 1988 on original units 3 and 4, and also shows the prototype performance expected from the 1958 model tests using a 2/3 Moody step-up. Units 3 and 4, although fabricated to the same design, were supplied by different manufacturers. All results are adjusted to a net head of 300 feet.

Homology of Upgraded Runner

Substantial effort was expended to confirm that the prototype upgraded runner was homologous to the model. To verify the model shapes, IMHEF made measurements of all principal runner dimensions, including blade profiles. Board section drawings prepared from these measurements were overlaid on computer design profiles to confirm the model-to-design homology. Use of the computer design profiles for prototype manufacture would therefore produce a prototype runner geometrically similar to the model.

A measuring frame employing full length blade templates was used to determine the final prototype blade profile. The blade templates were manufactured from computer generated design profiles and were spaced approximately 30 centimeters apart, and checked against mylar plots of the profiles. The blade profiles were then checked on grid of approximately 30 by 30 centimeters. Measurements by the Power Authority Quality Assurance staff confirmed that the final blade shape was more accurate than stipulated by International Electrotechnical Committee requirements.

During the runner fabrication process, blade angle, chords and vents were checked, including a final check after the last heat treatment and final machining step before the runner was shipped from the factory.

Results of Testing Upgraded Unit

Figure 5 provides the relative efficiency results of the acoustic-method field tests conducted in 1992 on the upgraded unit

4, and also shows the expected prototype performance from the 1989 model tests using a 2/3 Moody step-up.

Figure 5
Comparison of Acoustic Field Tests on Upgraded RMNPP Turbines and Expected Prototype Performance from 1989 Model Tests

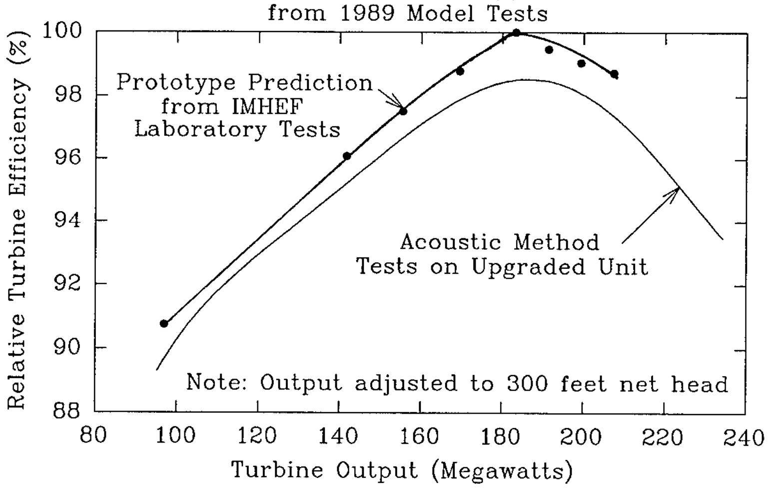


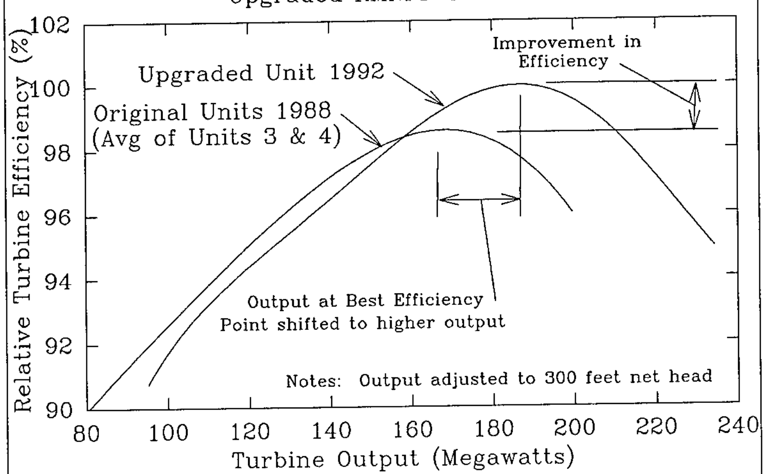
Figure 6 provides a comparison of the relative efficiencies from the 1988 field tests on the original turbines and 1992 field tests on the upgraded unit using the acoustic flowmeters. This figure indicates that an increase in the capacity at which peak efficiency occurs, and an increase in turbine efficiency, were achieved.

Load Rejection and Overspeed Tests

Load rejection and overspeed tests were conducted on the upgraded unit in 1992, but a full runaway test was not included. The purpose of these tests was to demonstrate that the upgraded unit could withstand the stresses and vibrations present during load rejection, to verify the governor closing time and to verify the overspeed switch settings and the operation of the overspeed switches.

Load rejections were performed at power levels of 56 MW, 114 MW, 170 MW, 217 MW and 224 MW. Turbine and generator shaft runout, turbine housing vibration, scroll case pressure and speed data were collected. Based on the data collected, overspeed switch settings were established slightly above the maximum speed expected during a normal load rejection.

Figure 6
Comparison of Acoustic Field Tests on Original and Upgraded RMNPP Turbines



Computer simulations were performed using data collected during the model runaway speed studies. These studies closely predicted the maximum speed the unit reached during the load rejection tests.

The synchronous speed of the units is 120 rpm, and is not being changed as part of the upgrade. The runaway speed, under the highest expected net head of 315 feet, is 208 rpm for the original units and 220 rpm for the upgraded units, based on the model tests.

Conclusions

The field tests conducted employed careful and redundant head, flow and power measurements. Good repeatability, generally within a few tenths of a percent for statistically drawn efficiency curves, using the acoustic flow measurement system has been shown to be achievable when extensive care and time are taken to assure calibrations of the head, power and flow measuring equipment.

The turbine output at which best efficiency occurs on the upgraded unit of about 186 MW (250,000 HP) agrees closely with that expected from the laboratory tests of about 183 MW (246,000 HP). For the original units, the model tests indicated that best efficiency would occur in the 157 MW (210,000 HP) to 160 MW (215,000 HP) range, and was observed in the field to occur at about 163 MW (219,000 HP).

The acoustic method field test results indicate that the prototype turbines, both original and upgraded, test at lower efficiency than predicted using the 2/3 Moody step-up. However, it was expected from the comparison of model tests of the original and upgraded turbines that an improvement in efficiency of approximately 1 to 2% would be achieved. The field tests demonstrate that within the test error band the upgraded turbine is more efficient than the original design by approximately the amount predicted.

It is currently planned that acceptance tests using the acoustic method will be performed on all future turbine upgrades.

Project Status

Further unit upgrades are planned once additional modifications on the upgraded generator are completed and successfully tested.

Acknowledgements

Laboratory and field testing of the extent discussed in this paper could only be performed with the cooperation of the staff of IMHEF, the Power Authority (Niagara Project and Headquarters staff), Voith Hydro, Accusonics and Parsons-Main, and the support of Power Authority Management, to whom we wish to acknowledge our thanks.

References

1. Knowlton, Robert J., and Peter W. Ludewig, "Potential for Upgrading the Robert Moses Niagara Power Plant," Waterpower '87, August 19-21, 1987, Portland, Oregon.

2. Howe, J., R. Knowlton, P. Ludewig and J. Schmieder, "Turbine Runner Procurement for the Robert Moses Niagara Power Plant,' Waterpower '89, August 23-25, 1989, Niagara Falls, New York.

3. Halas, R. N., "Acoustic Flowmeter/Efficiency System Installation at the Niagara Power Project," Waterpower '89, August 23-25, 1989, Niagara Falls, New York.

TURBINE UPRATING AND INCREMENTAL GAINS MADE WITH EACH CHANGE

Steven C. Onken, P.E.*

ABSTRACT

The Oroville-Wyandotte Irrigation District in an on going program of uprating, modernization, and automation of its existing hydro plants replaced the Francis Runner at its Forbestown Powerhouse in October 1989. The replacement runner was specified to improve efficiency and output without increasing flow through the unit.

Acceptance testing using a computerized acoustic flow meter system revealed that the runner did not meet the manufacturer's guaranteed efficiency and excessive powerhouse vibrations were experienced at wicket gate openings less than 90%. As a result of the District's studies and working with the turbine manufacturer the unit was tested with the turbine shaft plugged and then unplugged. A new longer nose cone was installed and tested. Finally a new set of wicket gates were installed and tested. The efficiency of the unit was tested with each improvement. The paper will present the gain or loss of each change made, the cost, and the operation of the unit. In addition greaseless wicket bushings were installed to eliminate the environmental problems of a pressurized grease system.

DESCRIPTION OF PROJECT

The South Fork Power Project (FERC No. 2088) is located on the South Fork Feather River in California about ninety miles north of Sacramento. It was

* Project Superintendent, Oroville-Wyandotte Irrigation District, P.O. Box 581, Oroville, CA 95965-0581

developed under a partnership with the Power Purchaser Pacific Gas and Electric Company and the owner Oroville-Wyandotte Irrigation District. Three powerhouses, Woodleaf at 55 MW, Forbestown at 34 MW, and Kelly Ridge at 10 MW were completed in 1963 and the fourth powerhouse Sly Creek at 12 MW was completed in 1983.

Woodleaf Powerhouse a pelton unit was uprated to 60 MW in 1985 with a new runner. Sly Creek Powerhouse a Francis unit was uprated to 14.2 MW in 1988 with a new runner which improved efficiency by 7% and also eliminated a severe penstock vibration problem. This penstock vibration problem and solution was presented at WaterPower 89 "Runner Induced Penstock Vibrations".

The third unit to be considered for uprating is Forbestown Powerhouse which consists of one Francis runner unit with 795 feet of net head and a flow of 551.5 cfs. Forbestown was first placed into operation in March of 1963 and Norman R. Gibson Company performed the original acceptance test. The unit produced 45,724 Hp or 34.2 MW at a flow of 551.5 cfs with a peak efficiency of 93.9%.

TURBINE MODIFICATION

In 1965 the turbine vent area was increased from 715.5 square inches to 789.6 square inches by cutting back the trailing edges of each bucket in order to pass additional water from the upstream plants. The unit produced 51,390 Hp at a flow of 607 cfs with a peak efficiency of 94.4%. Additional cooling was added to the transformer to handle the additional 3.4 MW in generation. The generator was originally over sized by the manufacturer for this installation.

The mild steel turbine runner experienced heavy cavitation and in 1974 was removed for extensive welding and grinding cavitation repair. When reinstalled the output of the unit dropped to 34.2 MW and was operated at that output until 1985. Cavitation repairs consisting of welding stainless steel and grinding back to the bucket contour were completed each year during shut down periods.

INSPECTION FOR UPRATING

With the uprating of Woodleaf Powerhouse in 1985 there was renewed interest in looking at the efficiency of Forbestown Powerhouse. The vent area of

the buckets was measured and the calculated total discharge area was found to be 700 square inches compared to 789 square inches that was accomplished in the modifications in 1965. The inspection revealed that the bucket discharge edges did not fit the original templates and that the buckets had been distorted by considerable welding and grinding. An attempt was made in 1986 to reshape the buckets at a cost of $35,000.00. The reshaping effort increased the unit output to 35 MW. It was felt that additional reshaping would not improve output and that a new runner was needed.

BASELINE TEST

A baseline test of the existing runner was conducted on March 23, 1989, using the Dye Dilution method and the Upf-1000 Panametric Ultra-Sonic Transient Time method to determine flow. The two methods were used to develop a cost and accuracy comparison for future testing. The flow obtained from the sonic method produced a very smooth and repeatable curve and was used to compute all turbine and unit efficiency data. The unit produced 48,624 Hp at a flow of 639 cfs and a peak efficiency of 84.3%. This test also indicated that the wicket gate leakage in the closed position was excessive at 21.4 cfs. A visual inspection revealed excessive erosion and cavitation of the mild steel wickets and stainless steel facing plates as the cause of the leakage.

NEW RUNNER INSTALLED

A new fabricated stainless steel runner and nose cone designed and manufactured by American Hydro Corporation was installed in October 1989. Due to casting problems the stainless steel wicket gates to be supplied by General Electric Company were not installed until October 1991. This schedule provided for an opportunity to verify each hydraulic component's contribution to the overall efficiency gain.

An efficiency test was conducted December 19, 1989, on the new runner. Using the same test methods as in the baseline test the unit produced 53,123 Hp at a flow of 665 cfs and a peak efficiency of 88.6%. This is an increase of 4,499 Hp and a 4.3% gain in turbine efficiency, but it did not meet the manufacturer's guarantee of 92% turbine efficiency. In addition the unit ran very rough in the 50% to 90% gate opening range with as much as 1 MW power surges. The manufac-

turer, American Hydro, and the owner, Oroville-Wyandotte, evaluated the unit and determined that a test should be run with the hollow turbine-generator shaft unplugged to vent air to the runner discharge area. A similar efficiency test was run on September 6, 1990. The unit produced 51,799 Hp at a flow of 655 cfs and a peak efficiency of 86.2%. The unit operated smoothly through all wicket gate openings but had lost 1,324 Hp and 2.4% efficiency.

DRAFT TUBE INSTABILITY

American Hydro felt that the rough operation of the unit in the 50% to 90% gate opening was due to draft tube surge caused by hydrodynamic instability in the swirling flow in the draft tube. Admission of air through the hollow shaft eliminated the instability but reduced the efficiency by 2.4%. Straightening vanes welded to the draft tube walls were considered but research indicated there would be approximately a 2% reduction in efficiency. American Hydro chose to redesign the existing blunt nose cone from 11.68" in length to 22.5" in length to try to stablize the swirling flow in the draft tube. The nose cone was installed and the unit tested but there was no reduction in the rough operation at the lower gate openings.

The six inch diameter hollow section of the turbine-generator shaft was plugged with a steel plate at the turbine runner and a two inch diameter opening cut to allow a reduced amount of air into the draft tube. This allowed sufficient air into the draft tube to smooth out the operation at the lower gate openings with no measurable reduction in horsepower or efficiency.

NEW WICKET GATES INSTALLED

The 18 existing mild steel wicket gates were replaced with stainless steel wickets machined to the original dimensions in October 1991. Using the same test methods as in the baseline test the unit was tested November 8, 1991. The unit produced 54,508 Hp at a flow of 662 cfs at a peak efficiency of 91.4%. The peak efficiency was within the accuracy range of the ultra sonic flow meter method, therefore, American Hydro met its guarantee.

Summarized below is the chronology of the test data:

Date	Max Flow	Max. Hp	Peak Efficiency	Comments
3/30/63	551.5	45,724	93.97%	Start Up Test
11/16/65	607	51,390	94.4%	Vent Area Modification
3/23/89	639	48,624	84.3%	Baseline Test
12/19/89	665	53,123	88.6%	New Runner (Shaft Plugged)
9/6/90	655	51,799	86.2%	Shaft Unplugged
11/8/91	662	54,508	91.4%	New Runner, New Wickets, New Nose Cone

The wicket gate leakage in the closed position was reduced from 21.4 cfs to 8.6 cfs with the installation of the new stainless steel wicket gates and facing plates.

GREASELESS WICKET BUSHINGS

During the replacement of the wicket gates it was found that the conventional pressurized greased bronze wicket bushings were not being lubricated properly and excessive wear was occurring. Based upon an independent study for British Columbia Hydro a Thordon SXL elastomer bearing was selected to replace the wicket gate bushings and linkage bushings thus eliminating the maintenance of a complex pressurized grease system to 100 lubrication points. Total cost for the 54 wicket bushings and 36 linkage bushings was $32,000.00. The elastomer bearing will assure proper lubrication and eliminate the environmental problems of the waste grease from a pressurized grease system.

CONCLUSION

What was thought to be a simple uprating of a Francis unit with a new runner turned into a complex three years of persistent effort to achieve the desired

results. It took a team effort of the owner and the turbine manufacturer working together to identify the problems, evaluate alternatives, and test the modifications made. The results are summarized below:

Description	HP	Flow cfs	Eff.	Total Cost	First Year Benefit
Baseline Test	48,624	639	84.3		
New Runner	53,123	665	88.6	$313,000	
Gain	4,499	26	4.3		$268,300
Wicket Gate	54,508	662	91.4	$324,000	
Gain	1,385	-3	2.8		$174,700
Total Gain	5,884	23	7.1		$443,000

The experience gained from this uprating has been invaluable for Oroville-Wyandotte and the power purchaser, Pacific Gas and Electric Company. The efficiency testing using the inexpensive ultra sonic flow method allowed the determination of each incremental change in the turbine. When the runner did not meet the manufacturer's guarantee it was assumed the manufacturer had failed but the testing and replacement of the wicket gates confirmed that American Hydro had designed and built a replacement runner that met their guaranteed performance.

REFERENCES

Steven C. Onken "Runner Induced Penstock Vibrations" 1989 WaterPower Volume III Page 1328.
H.T. Falvey 1971 "Draft Tube Surges" Hydraulics Branch; Division of General Research; Engineering and Research Center; Denver, Colorado.

COMPARISON OF MAXIMUM EFFICIENCIES IN A CROSS–FLOW TURBINE

By

V. R. Desai[1], and N. M. Aziz[2], Members, ASCE

ABSTRACT

This paper compares the maximum experimental efficiency in a cross–flow turbine with that obtained by the linearity assumption between the volume of entrained water in the runner with time, as torque increases.

These comparisons are made for 33 cases involving various geometric and operational parameters such as the angle of water entry into the first stage inlet of the turbine, the runner inner–to–outer diameter ratio, the number of blades, and the mode of turbine speed control. The results indicate that the linearity assumption between the shaft speed and torque is extremely fast, but somewhat inaccurate in predicting the maximum efficiency.

NOMENCLATURE

D_1 Runner Outer Diameter (m)
D_2 Runner Inner Diameter (m)

[1]Doctoral Student and [2]Associate Professor, Department of Civil Engineering, Clemson University, Clemson, South Carolina 29634–0911.

H	Head (m)
I	Moment of Inertia of the Runner (kg.–m^2)
N	Shaft Rotational Speed (rev./min. i.e., rpm)
n_b	Number of Blades
N_0	Shaft Runaway Speed (rpm)
P_{in}	Input Power (watt)
P_{out}	Output Power measured at the Shaft Center (watt)
Q	Flow Rate (m^3/sec. i.e., cumec)
T	Shaft Torque (N.–m.)
α	Angular Acceleration (rad/sec^2)
α_1	Angle of Attack for the Water flowing (deg.)
γ	Specific Weight of Water (N/m^3)
η	Efficiency (%)
ω	Angular Speed (rad/sec.)
ω_0	Runaway Angular Speed (rad/sec.)

INTRODUCTION

With the increase in the awareness for environmental conservation, renewable energy sources such as hydropower are attracting more attention than the sources like nuclear energy, coal, and oil. In hydropower, low–head projects are preferred to either the medium–head or the high–head projects, due to their inherent advantages of low cost. However, there are only few turbines specifically designed for low–head hydro. The cross–flow turbine (CFT) is specifically conceived for low–head hydro and therefore occupies a prominent position among the low–head hydropower turbines.

The cross–flow turbine is extremely simple in structure and has a great potential to adjust itself to flow fluctuations. The two main components of the cross–flow turbine are the runner and the nozzle. The runner has at least two circular side walls with the blades connecting these side walls. The nozzle is rectangular in cross–section and is responsible for directing the flow into the runner at a specified angle of attack (α_1), as shown in Figure 1. The water jet can hit the blades inside the runner either once or twice depending upon the proximity of its emerging point in the nozzle to the nozzle

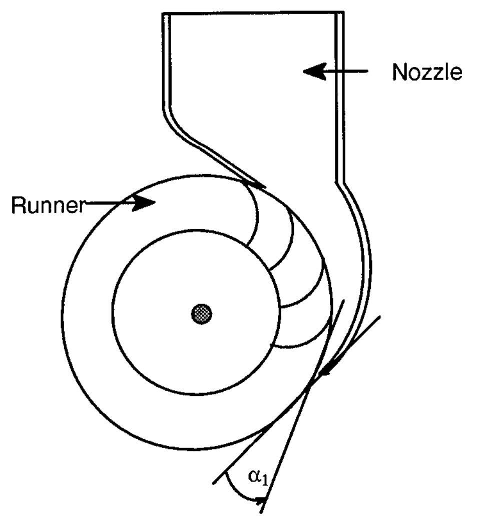


Figure 1. Definition Sketch of the CFT

tip. That portion of the jet which hits the blades twice is known as the 'cross–flow', and hence the name of the turbine. Figure 2 shows a typical flow pattern in a CFT.

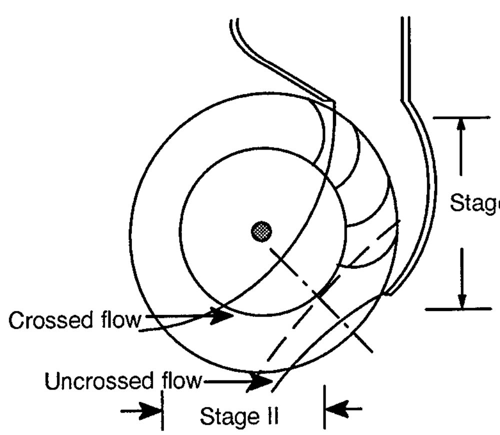


Figure 2. A Typical Flow Pattern in a CFT

In the past, there have been several attempts to analyze the CFT performance theoretically as well as experimentally. The reader is referred to Desai and Aziz (1992) for a brief literature review on CFT.

THEORY

From the principles of dynamics, the expression for the torque (T) acting on the CFT can be written as follows:

$$T = - I.\alpha \tag{1}$$

where the negative sign indicates deceleration. If the angular speed of the turbine is varied uniformly in t seconds from an initial (i.e., when the torque is zero) run–away speed (ω_0 radians/sec.) to a particular angular speed (ω radians/sec.), Equation (1) reduces to:

$$T = \frac{I}{t}.(\omega_0 - \omega) \quad . \tag{2}$$

The moment of inertia (I) of a CFT runner is a function of the entrained water jet and the runner geometry. The water jet volume entrained inside the CFT runner varies as the runner speed and will be maximum when the runner speed is zero. Thus the moment of inertia varies from an initial value at run–away speed to a maximum value at zero speed. The contribution of the water jet to the moment of inertia can be approximated by dividing the flow into six regions, as explained by Fukutomi et al. (1991). However, this is a very tedious procedure requiring significant computer time and memory. So, a simple alternative procedure is explained here in the following paragraphs.

If the flow rate (Q) and the operating head (H) are maintained constant, the run–away speed (ω_0) will also be a constant. Further, if we consider the ratio I/t to be constant, Equation (2) can be solved for ω.

Expressing the angular velocities ω_0 and ω in terms of the initial run–away rpm (N_0) and the rpm at torque T (i.e., N) respectively, equation (2) simplifies to:

$$T = \frac{I}{t} \cdot \frac{2\pi}{60} \cdot (N_0 - N) \quad , \tag{3}$$

i.e.,
$$N = N_0 - \frac{t}{I} \cdot \frac{30}{\pi} \cdot T \quad . \tag{4}$$

For the CFT efficiency (η) to be a maximum at a given flow rate (Q) and head (H), the Output Power (P_{out}) should be a maximum. P_{out} is expressed as:

$$P_{out} = T.\omega \quad , \tag{5}$$

or,
$$P_{out} = T.\frac{2\pi N}{60} \quad . \tag{6}$$

For maximum power, the product N.T must be maximum, or in other words

$$\frac{\partial[N.T]}{\partial N} = 0 \quad . \tag{7}$$

Substituting the value of T from Equation (3), Equation (7) can be simplified to:

$$N = \frac{N_0}{2} \quad . \tag{8}$$

Hence, according to the analysis outlined above, the maximum efficiency should occur at one half the run–away speed based on the linearity assumption between the shaft torque and speed of a CFT. The implications of this assumption are studied by fitting a cubic curve for the same sets of torque and speed data obtained for a CFT. Here the cubic curve is selected because of the following reasons: it has been observed that polynomials having an odd number as their order, satisfy the end conditions of the torque–speed data reasonably well. A cubic curve is the next higher

odd–ordered polynomial after the straight line, and hence the selection.

DATA ANALYSIS

The performance of turbines is measured by their efficiencies. In general, efficiency is an indication of what percentage of the input to the turbine is converted into power output. Efficiency (η) is defined as:

$$\eta = \frac{P_{out}}{P_{in}} 100\% \tag{9}$$

where, $$P_{in} = \gamma HQ \quad . \tag{10}$$

The straight line equation for the torque–speed data will be in the form of Equation (4), which can be re–written as:

$$N = N_0 - b.T \quad . \tag{11}$$

Maximum efficiency is achieved when Equation (8) is satisfied, and the corresponding torque is:

$$T = \frac{N_0}{2.b} \quad . \tag{12}$$

These values of speed and torque as given by Equations (8) and (12) can be substituted in Equations (6) and (9) to obtain the maximum efficiency based on the linearity assumption.

The cubic curve–fit equation for the torque–speed data is of the form:

$$N = a_0 + a_1.T + a_2.T^2 + a_3.T^3 \quad , \tag{13}$$

where a_0, a_1, a_2, and a_3 are coefficients. 'Tablecurve', a curve fitting software package, was used to fit the cubic curves for these sets of data. By applying

the condition stipulated in Equation (7), the values of torque and speed at the maximum efficiency point were determined. The roots of the cubic polynomial resulting from Equation (7) were obtained by using 'Matlab', a mathematical software package.

RESULTS

Table 1 lists the maximum experimental efficiency as reported by Aziz and Desai (1991), maximum efficiency based on the linearity assumption, and the maximum efficiency based on the cubic curve fit. The comparison between the maximum experimental efficiency and maximum efficiency (linear fit) indicates that, in 25 out of the 33 cases the maximum efficiency (linear fit) is higher but the difference is always within 5.6%. This difference has a mean value of 1.6% and a standard deviation of 2%.

Similarly the comparison between the maximum experimental efficiency and the maximum efficiency of the cubic equation shows that, in 21 out of 33 cases the maximum efficiency (cubic fit) is higher than the experimental value but the difference is always within 2.1%. This difference has a mean value of 0.2% and a standard deviation of 0.9%.

Figure 3 is a plot of experimental torque versus speed data along with the fitted equations. Likewise, Figure 4 shows the efficiency plots for the same data. The data plotted in Figures 3 and 4 are for the worst case.

CONCLUSIONS

Both linear and cubic curve–fits can eliminate the point uncertainties in the experimental measurements. However, linear curve–fit is very easy to incorporate. It is extremely fast in the preliminary estimation of the maximum efficiency.

The validity of the linear fit is dependent on the validity of the assumption that I/t is constant. It is easily observed in the laboratory that, as the torque and speed change the volume of the water entrained within the runner changes. However, it needs to be determined whether this change makes the moment of inertia a linear function of time.

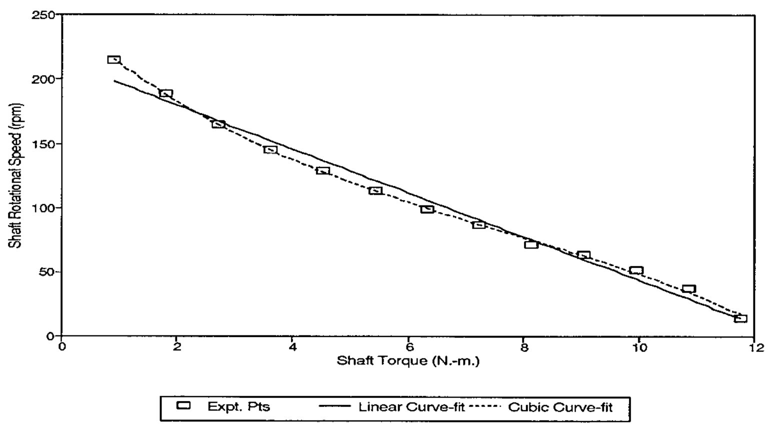


Figure 3. Torque–speed Plot along with the Curve–fits

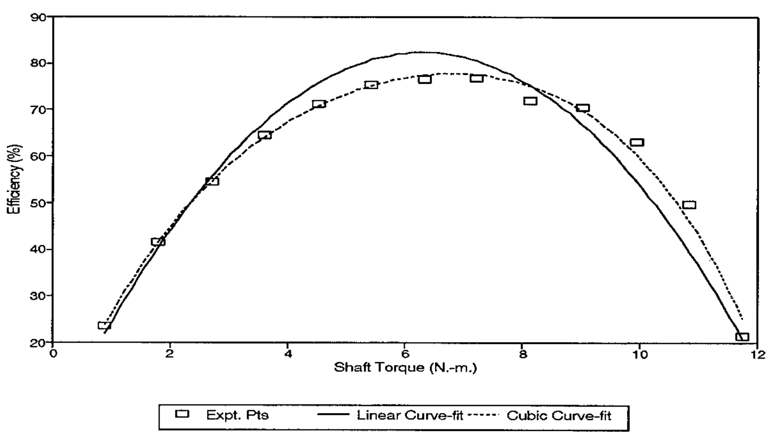


Figure 4. Efficiency Plots for the Same Data as in Figure 3.

Table 1. Runner Parameters and Efficiencies

Runner #	Angle of Attack $\alpha°$	Dia. Ratio D_2/D_1	# of Blades N_b	Speed Contrl. Mode	Max. Exptl. Effic. $\eta_{max\ exp}$ %	Max. Effic. (linear) $\eta_{max\ the}$ %	Max. Effic. (cubic) $\eta_{max\ cub}$ %
1	24	0.60	15	Auto.	64.1	67.2	64.5
2			20	Auto.	64.4	66.1	65.1
3			25	Auto.	74.5	79.6	74.0
				Manual	80.6	86.0	81.4
4		0.68	15	Auto.	59.1	56.7	57.5
5			20	Auto.	70.7	75.4	70.9
6			25	Auto.	76.7	82.3	77.7
				Manual	84.5	89.0	85.7
7		0.75	15	Auto.	50.6	50.2	50.2
8			20	Auto.	63.3	61.8	61.4
9			25	Auto.	71.9	71.2	71.1
10	28	0.60	15	Auto.	63.7	64.8	63.6
11			20	Auto.	71.1	72.8	70.8
12			25	Auto.	71.7	74.0	70.0
				Manual	78.0	79.6	78.5
13		0.68	15	Auto.	56.5	57.6	57.4
14			20	Auto.	65.2	66.2	64.8
15			25	Auto.	66.7	69.3	67.2
				Manual	74.0	75.9	74.1
16		0.75	15	Auto.	49.8	49.7	50.6
17			20	Auto.	59.6	59.0	60.6
18			25	Auto.	65.2	65.9	64.6
19	32	0.60	15	Auto.	58.3	57.5	57.8
20			20	Auto.	64.8	67.6	65.4
21			25	Auto.	66.3	69.0	66.1
				Manual	73.7	74.9	74.1
22		0.68	15	Auto.	55.6	54.9	55.2
23			20	Auto.	63.1	63.9	63.2
24			25	Auto.	64.3	67.0	64.9
				Manual	71.0	72.7	73.1
25		0.75	15	Auto.	46.4	47.1	48.5
26			20	Auto.	55.1	57.1	56.5
27			25	Auto.	62.7	64.6	63.0

ACKNOWLEDGEMENTS

The funds provided by the U. S. Department of Energy, Bonneville Power Administration, Portland, Oregon, enabled the authors to perform this research.

APPENDIX: (REFERENCES)

Aziz, N. M., and Desai, V. R., (1991), "An Experimental Study of the Effect of some Design Parameters on Cross–flow Turbine Efficiency", Engineering Report # 1W–91, Department of Civil Engineering, Clemson University, Clemson, South Carolina 29634–0911

Desai, V. R., and Aziz, N. M., (1992), "An Experimental Investigation of Cross–flow Turbine Efficiency", Hydropower Fluid Machinery, the Winter Annual Meeting of the American Society of Mechanical Engineers, Anaheim, California, November 8–13, pp. 7–14

Fukutomi, J., Senoo, Y., and Nakase, Y., (1991), "A Numerical Method of Flow through a Cross–flow Runner", JSME International Journal, Series II, Volume 34, No. 1, February, pp. 44–51.

MINGTAN - HIGH HEAD REVERSIBLE PUMP/TURBINE

Robert D. Steele, P. E.*

Project Overview

Taiwan Power Company (Taipower) is the sole power utility established by the government of Taiwan, ROC. The Mingtan project was the second major pumped-storage plant developed by Taipower for the island of Taiwan. At 1,600 MW (6 units rated at 275 MW) the Mingtan Pumped Storage Project has the largest installed capacity of any pumped-storage project in Southeast Asia. The powerhouse was constructed in an underground cavern. The existing Sun Moon Lake serves as the upper reservoir for the project. The lower reservoir was formed by the construction of the Shuili Dam, 400 m below the headwater of Sun Moon Lake.

Project Characteristics

No. of Units	6
Design Head	590 m
Turbine Capacity	312 MW
Pump Capacity	275 MW
Turbine Head	401 m
Pump Head	411 m
Synchronized Speed	400 RPM

Introduction

The design of the 6-275 MW pump/turbines for the Mingtan Project was influenced by the demand for superior hydraulic and mechanical performance and long term reliability. The design considered the severe

* Manager, Engineering, Voith Hydro, Inc., P.O. Box 712, York, Pennsylvania 17405, Telephone: (717) 792-7000

operating conditions associated with typical pump-turbine operation, such as frequent turbine starting and stopping, synchronous condensing operation, pump starts, pump power failures, turbine load rejection, and the pressure and speeds associated with these operations.

Mechanical Design Considerations

The designer of a high head pumped-turbine must consider stress levels and deflections, while recognizing that the long term reliability is influenced by vibration amplitude and frequency. The high capacity, high head, high speed pumped-turbine design is a result of a thorough analysis of the mechanical components that form the main structure. Hydraulic performance is dependent on the close running clearances between rotating and stationary components and structural members with shape and thickness that promote performance. The design of a high performance pump-turbine requires a study of the structure using finite element methods (FEM). These studies can predict the area of high stress, high deflection and can be used to identify areas that should be investigated for fatigue life studies. Included in these studies are vertical deflection of the head cover to check on rigidity, angular rotation of the top deck of the headcover to check for wicket gate binding, radial growth of rotating components to check for wearing ring clearance, and deflection in areas of O-ring seals to check for seal extrusion.

Throughout the design of Mingtan, as the structural components were optimized mechanically and hydraulically, it became more evident to the designer that material selection would be a primary factor in the design. Reviewing the design of the main structural components (foundations, discharge ring, stay ring, spiral case and headcover), it could be seen that enhanced material properties would be a very important factor in providing equipment to satisfy design requirements. A review of each of these components and the effect on the total structure follows in a detailed review of each component.

Foundation

The discharge ring was designed to be the structural component on which the remainder of the unit would be constructed during field erection. That is, during site erection, the unit level and center would use an embedded discharge ring as the control for erection. A completed stay ring/spiral case could then be totally supported by a discharge ring during erection, pressure testing, and embedment, and there would not be the necessity of supports on the stay ring or spiral case. The bolted joint between the stay ring and discharge ring would complete the mechanical tie to the structure and the tie to the foundation would use the extension of

the same bolts to transmit tensile loads deep into the powerhouse concrete (see Figure 1). This design would allow the stay ring/spiral case components to be set using machined surfaces for support, then the bolts holding the discharge ring to the stay ring could be pre-stressed to values predicted by a FEM Stress Analysis. After the entire discharge ring, stay ring, spiral case and pit liner are embedded and grouted, the bolts holding the structure to the powerhouse concrete could be preloaded to values predicted by the FEM Structural Analysis.

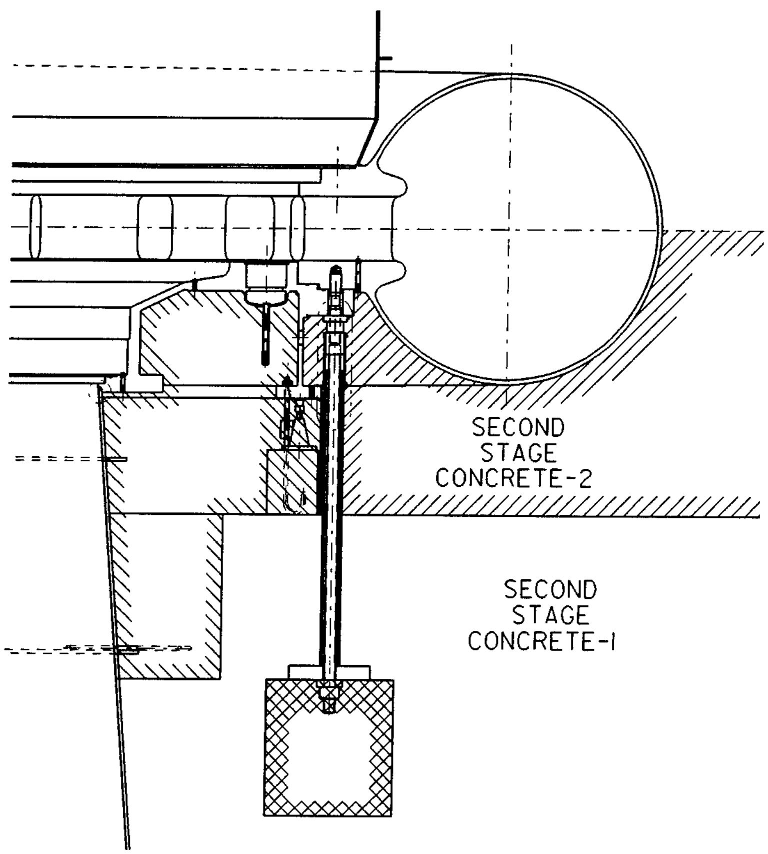


Figure 1

Stay Ring/Spiral Case

In order to meet the very quick delivery, the field installation schedule for the Mingtan Project required the manufacturer of the equipment to use components fabricated from plate steel. In the design of the stay ring/spiral case, the FEM of the stay ring and stay vanes showed the tensile loads required the material to be an alloy steel. For the project, the material HY-80 was selected (yield strength = 550 N/mm^2) for the stay vanes. (This allowed the thickness to be minimal to satisfy hydraulic requirements) These stress levels in the vanes required the use of high yield strength steel plates for the upper and lower stay ring decks. The intersection of the vane and deck required a special fillet weld to distribute the stress from the vane (yield strength = 550 N/mm^2) through the deck (yield strength = 260 N/mm^2). These special fillets were constructed using weld overlay and were hand ground to templates.

The spiral case was fabricated using high yield material and the deck of the stay rings were sloped so that the intersection would produce an optimum weld joint. The materials selected for the spiral case plates were HY-100 (yield strength = 690 N/mm^2) and these plates were welded to the stay ring deck material (yield strength = 260 N/mm^2). This critical intersection of the stay ring and spiral case was modeled and studied during the FEM Analysis, and then the weld intersection was selected to produce a constant stress fillet again produced by welding and hand grinding to templates. For the Mingtan project, the unit size allowed the spiral case to be shop welded to the stay ring, stress relieved and machined prior to shipment. This shop welding allowed the field erection time to be reduced which supported the site installation sequence.

Headcover

It was stated earlier that the deflection of the components must be studied to optimize the location of the bolted joint between the headcover and the stay ring. The placement of the bolt circle diameter and bolt configuration had been studied to reduce the stress impact on the vane tensile stress, and the bolting flange location was placed at the lowest elevation possible to the stay ring deck surface to reduce the deflection of the head cover under pressure. The effort to reduce deflection relates to the minimal clearance acceptable on the impeller wearing rings, and the vertical movement associated with headcover rotation relates to the control of guide bearing clearances. A review of stress and deflection levels of the structure shows a necessity to use a material with higher yield strength to maintain a contractually specified factor of safety.

Discharge Ring

In many pump-turbine designs of the past, the units were supplied with a discharge ring and a bottom ring as a separate removable component. For Mingtan, the bottom ring and discharge ring were combined into a integral unit to improve the structural rigidity. The structure was checked to confirm foundation loads, the gate stem loads, and the rotation of the structure when pressure was applied. The Mingtan unit has gate end seals, and this check on the movement of the surface under the gate shows the impact on wicket gate leakage during condensing operations and pump starts.

In a review of the stress and deflection through the FEM study, the necessity to use materials with higher yield strengths for the ribs is shown to keep the factor of safety within the specified limits.

Wicket Gates

For a high head pump-turbine, the design of the wicket gate is influenced by many considerations. For example, stress, deflection, deflection of the adjacent components, clearances in guide bushings, frequency, and the impact these criteria have on the fatigue life, must be considered. These factors tend to increase gate stem size and leaf thickness, which impact on pump/turbine performance. For Mingtan, a forged stainless steel material was selected, (A182 F6NM), because a forging has better grain structure, its as-forged shape is near design, the shape is consistent from gate to gate, the forging has excellent toughness for a fatigue environment, and its strength is consistent with project requirements.

The wicket gate loads due to water pressure and component weight are supported by self-lubricated bushings. For the Mingtan Project, the Oiles Bearing 500 SP was selected to serve as the bushing that supports the gate stem hydraulic loads. These bushings have solid graphite composite lubricant buried in the bronze base metal. This type of bushing on a pump/turbine, must be protected from dirt by sealing the stem using a pressure activated seal. The linkage bushing for the gate mechanism was also selected to be an Oiles Bearing. The maximum load on these bushings was specified to be no more than 250 Kg/cm^2.

Self-lubricating thrust bearings were also supplied for the gate mechanism to take the weight of the components, and to take the hydraulic thrust of the gate stems. For Mingtan, Lubron Bearings were selected for the project and these bearings were also lubricated with a compound that includes graphite.

The wicket gate mechanism was provided with a shear pin to protect the mechanism from overload from gate blockage, and a movement restraining device to prevent rapid movements of the gate arm caused by hydraulic torques should the shear pin fail. Loads associated with the shear pin are part of the analysis to confirm wicket gate strength. The device to prevent rapid movement was included as a feature of the gate arm and shear lever design. The design included a bronze sleeve on the arm, clamped with a preloaded fastener on the shear lever.

Impeller

The impeller, cast of A487 CA6NM (13Cr-4Ni), was reviewed for its deflection under overspeed conditions where particular attention was given to the clearance between the rotating and stationary wearing rings. In general, the stresses in the impeller were low for the material (yield strength = 515 N/mm^2), however, the intersection of the vanes, crown, and band required a review and where vane edges were thin for hydraulic performance, the loading was reviewed to prevent possible fatigue damage. For Mingtan, the wearing ring surfaces of the impeller are an integral part of the impeller casting.

Mechanical Shaft Seal

A mechanical hydrostatic face seal was supplied to seal the shaft where it passes through the headcover. The face seal was selected because it allows the shaft to have radial movement without damage to stationary components, and it has multiple springs in the seal follower that preloads the seal faces together. This type of seal was selected because it uses clean water injected in the seal faces to:

- Produce the lowest possible wear rate
- Prevent silt laden water from entering the seal faces
- Keep cooling water on the seal faces knowing the shaft speed at the seal would sling the water away from the seal faces
- Lubricate the seal faces, as the unit will run in air as a synchronous condenser and during pump starts

Guide Bearing

As stated earlier, it was very important to construct the bearing housing that could not be structurally influenced by the rotation of the headcover or the main shaft load supported by the bearing shoes. For

Mingtan, a multiple shoe bearing was supplied. The oil selected to be used in all bearings was ISO 46, and therefore for that viscosity oil, a twelve shoe bearing assembly was supplied. The space allocation for the guide bearing in a high head pump/turbine is limited, and for Mingtan the guide bearing sump was located outside the pump/turbine pit with pumps and oil coolers located on the sump.

Stationary Wearing Rings

Wearing rings provide the close running clearances between the stationary components and the rotating impeller. These stainless steel wearing rings were supplied in Nitronic 60, in an effort to minimize the risk of seal damage should the impeller touch the wearing rings. The clearance on these wearing rings was carefully established using data from the model.

Installation

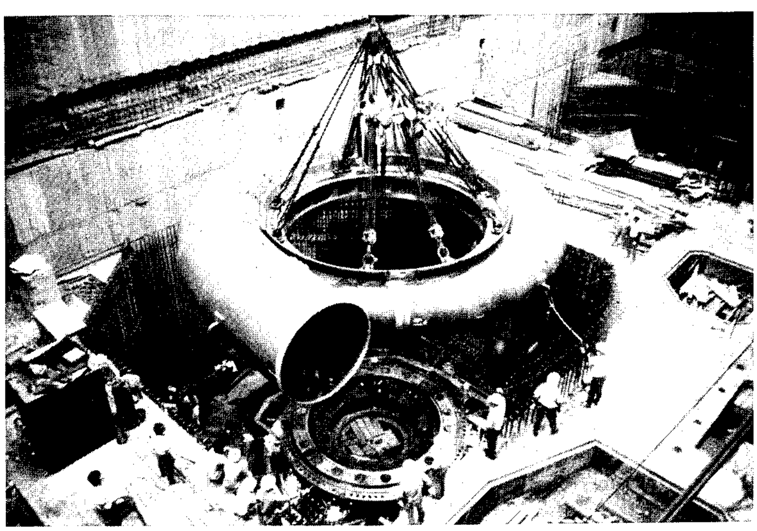


Figure 2

The pump/turbine design for Mingtan was reviewed for its adaptability to the site erection sequence. In the early design decisions, it was apparent that the discharge ring had to be the foundation for the total pump/turbine unit and this component was the first to be erected. The stay ring, spiral case and spiral case extension were shipped in five sections,

with joints prepared for field welding of the spiral case joints and the stay ring cross section.

While the stay ring/spiral case sections were welded together, the powerhouse construction continued with the discharge ring installation and embedment (see Figure 2). This technique allowed the erector to install all embedded piping and conduit under the spiral case, stay ring and discharge ring before the spiral case was installed. With the spiral case and stay ring joints welded and Nondestructive Examination complete, the assembly was field machined to establish critical turns, bores, elevations and flatness. In the site erection sequence, one week was allocated for this machining operation. Once machined, the entire assembly (weight approximately 85 metric tons) was lifted from the erection floor (see Figure 2) and placed on the embedded discharge ring. The stay ring, spiral case and spiral case inlet were set to centerline and rotation, and all bolting was then pre-stressed. This erection sequence allowed the assembled unit to be pressure tested with the completed unit sitting in a permanent condition, then the embedment could continue.

With the unit assembled, embedded and grouted, it was planned to then finish bore the gate stem holes in the discharge ring. It had been planned to use the centerline of the headcover to establish the true centerline of the gate stem holes, so that the discharge ring holes could be in exact location by line-boring with portable field machining equipment.

Conclusion

The trend to increase unit capacity and design pressure for pump/turbine projects demands that manufacturers provide the analyses that can predict component behavior under various design conditions. The design tools that are available today provide a significant contribution to the development of hydropower technology that can optimize performance and increase reliability. The assumptions made during the design of a high capacity pump/turbine must be monitored throughout the design, manufacture, and installation phase to be sure that critical stress areas, manufacturing tolerances, installation procedures, alignment tolerances, preloads, etc., have been executed to exacting tolerances while producing a completed project that meets the expectations of the end user.

BAD CREEK & MINGTAN - PUMP/TURBINE COMMISSIONING

Laurence F. Henry *

Abstract

Being literally a world apart in location and culture, the Bad Creek and Mingtan pumped storage projects each produce about 1600 MW of electrical power by utilizing multiple, high head, single stage, reversible pump/turbines in an underground powerhouse.

During commissioning pertinent data channels were recorded on a visacorder for immediate review.

For permanent records in one plant, analog magnetic tape recording was performed with analysis being done later by use of a main frame computer. At the other plant recording and analysis were done on a personal computer (PC) system.

On both projects the "slow-start" technique for accelerating the unit from at-rest to rated speed conditions was utilized. Data comparing this method to the "quick-start" generally used on turbines will be shown.

The desirability of special tests for pump priming will be covered with comment on the technique needed for safe use in the prototype. Five different methods of sensing the "primed" pump condition will be mentioned.

From the aspect of the pump/turbine machinery, the favored situation for mechanically balancing of the rotating mass entails accelerating the mass up to speed with the electrical starting system while the impeller (runner) rotates in air or compressed air. Comments on the balancing method will be offered.

* Consultant, Voith Hydro, Inc., P.O. Box 712, York, Pennsylvania 17405, Telephone: (717) 792-7000

Introduction

Bad Creek, located in the United States in South Carolina and operated by the Duke Power Company, contains four 360 MVA units. Mingtan, located in the central highlands of Taiwan and operated by the Taipower Company contains six 300 MVA units.

For pump starting Bad Creek uses auxiliary pony motors while Mingtan has a static frequency converter; but both projects rely on back-to-back synchronous starting with one of the other pump/turbines for additional starting security.

These projects abound with similarities including the fact that each utilizes an existing reservoir in its scheme. The lower reservoir for Bad Creek existed as the upper reservoir for the Jocassee Pumped Storage Station. The upper reservoir for Mingtan is Sun-Moon Lake which also serves as the headwater for the Minghu Pumped Storage Station.

TABLE 1

MINGTAN		BAD CREEK
6	Number of Units	4
300	Electrical Rating Generator MVA	360
283	Motor MVA	354
400	Rotational Speed - RPM	300
380	Rated Generating - Head - Motors	354
275	Rated Power - MW	366
410	Rated Pumping Head - Meters	379.5
52	Rated Pump Flow - cms	72.5
4250	Impeller Diameter - mm	5319
7 20	Impeller Vanes Wicket Gates	7 20
470	Distributor Height - mm	480
965	Shaft Diameter - mm	1118
2300	Main Isolation Valve Dia. - mm	2566

Differences in the commissioning programs and sequences varied due to the existence of the headwater reservoir at Mingtan, compared with the necessity of storing water in the upper reservoir at the Bad Creek captive pumped storage location.

This dictated operation in the pumping mode first for Bad Creek, while allowing the choice of the generating mode of operation at Mingtan.

General information for both projects is shown in Table 1, but only the Mingtan distributor section is shown in Figure 1.

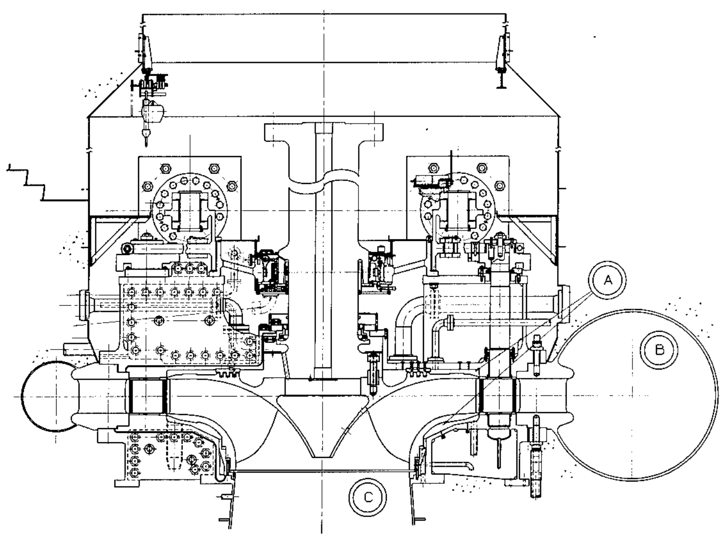


Figure 1

Data Acquisition

At Bad Creek, on site monitoring during commissioning was accomplished with a fiber optic visacorder. Portable spectrum analyzers provided frequency information as required on site. For massive data storage a 28 channel analog magnetic tape recorder was used in parallel with the visacorder with analysis being done later by use of a mainframe computer. This highly successful technology has been used for more than 20 years in the commissioning of hydro machinery.

At Mingtan the visacorder was also used on-site with mass storage being done through a personal computer (PC) system. This capitalized on the turbine manufacturer's experience with computer, data acquisition and digital techniques. Versatility is the reason for use of a PC since the PC can store, recall, analyze, plot and tabulate data with the objective of issuing a report in the expeditious manner.

The data system is configured with a 386-20 megahertz (MHz) laptop PC with a 100 megabyte (MB) hard drive. An expansion box with a 16 bit 64 channel analog to digital conversion capability, allows 32 active data channels to be scanned at a frequency of up to 2500 Hz per channel. The maximum scan frequency changes as the number of active channels is changed, but this frequency suits the task of monitoring hydro machinery very well.

The system interfaces with a micro cassette tape drive for mass data storage. The transfer rate of data exceeds 1.5 MB per second with this device. This allows long transient tests as well as steady state test data to be captured during unit commissioning.

Although the system utilized existing hardware, the sophisticated software was written in-house. The system innovator described the hard drive transfer rate and the software as the key to the use of a PC system for such varied field use.

As an analyzer, the system can give spectral data, along with time-amplitude plots and statistical deviation for test uncertainty. Index, pressure-time and volumetric testing programs have been perfected with an entire package for in-plant monitoring, trending and setting thresholds soon to follow. A hydraulic transient program gives the commissioning engineer on-site capability to study hydraulic transient phenomenon if the need arises.

Slow-Start

Inherently high head reversible pump/turbines are slightly more difficult to synchronize to the electric grid than a conventional turbine for the same conditions. Wicket gate maneuvers to adjust unit rotational speed also cause pressure variations in the water passages which further exacerbate speed changes and synchronization. For this reason a "slow-start" regime has been used on the Bad Creek and Mingtan units.

Figure 2 contrasts the pressures variations for a typical high head pump/turbine with a long penstock, but no surge tank, for "fast" and "slow" wicket gate opening rates.

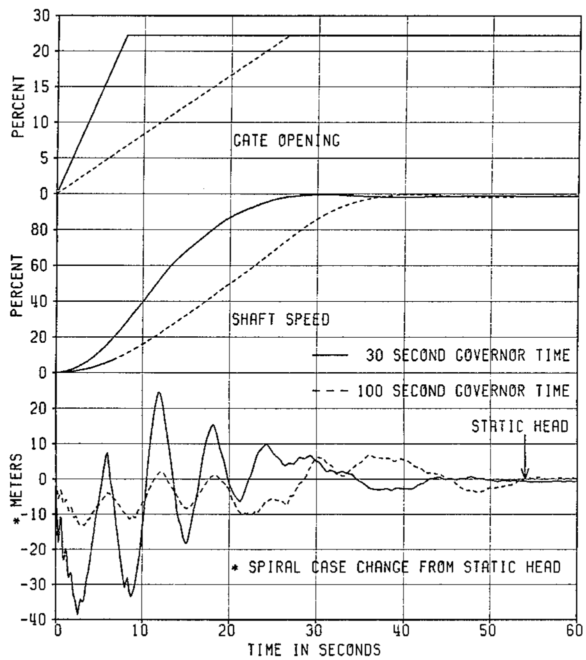


Figure 2 - Turbine Start

The pressure variations depicted in Figure 2 for the spiral case pressure are large and stem from the inability of the water to accelerate quickly from rest when the wicket gates are opened at a 30 second nominal rate which would be common for this type of machine.

The pressure variations for a "slow-start" using a 300 second nominal wicket gate opening rate as shown in the figure are much less severe. This translates into a stable rotational speed, devoid of slight speed variations, at the time when the wicket gates attain the speed-no-load position.

For "fast-start" the wicket gates rapidly attain a larger than speed-no-load position and hold until the governor senses 95 percent of synchronous speed and repositions the gates to hunt for stable synchronous speed.

Conversely for a "slow-start" the wicket gates open slowly so that stable speed-no-load conditions are reached without gate hunting and subsequent penstock pressure variations.

Since a high pressure thrust bearing oil lift system is used on these and most other modern hydro machines, there is no danger of thrust bearing damage associated with the "slow-start" procedure.

Slower wicket gate opening also minimizes the draft tube pressure rise which helps the main shaft seal to perform under less extreme dynamic conditions.

In practice the "slow-start" procedure works well and test data indicates that shafting system vibrations especially in the axial direction are minimized. Elapsed time from at rest to synchronized frequently favors the "slow-start" because the pressure and speed variations at rated conditions are smoother.

Based upon personal observations many conventional hydro turbines could benefit from a "slow-start" procedure especially with respect to wear on the machinery. Benefits can be documented by the use of modern test instrumentation in a simple test program on most turbines.

Pump Priming

Some methods for sensing the pump "primed" condition for gated reversible pump/turbines are:

1. Pressure in the impeller chamber - the area between the impeller (runner) seal and the closed wicket gates. See Figure 1 Location A.

2. Differential pressure between the impeller chamber and the penstock or volute (spiral case). Location A to B.

3. Motor power input.

4. Suction tube (draft tube) pressure and presence of water in compressed air vent line. Location C.

5. Differential pressure between impeller chamber and suction tube. Location A to C.

Some plants combine two of these sensing methods which can lead to inconsistent priming practice.

For the Mingtan upper reservoir with daily fluctuations of 1.2 meters, the single pressure sensor (Item 1) is quite effective.

For the Bad Creek upper reservoir weekly fluctuations of 50 meters, the differential pressure sensor (Item 2) allows the optimization of the "primed" condition regardless of the reservoir water elevation.

Optimization of the pump "primed" condition refers to opening the wicket gates to establish pump flow at just the right instant to accomplish the safest, smoothest, most vibrationless operation. Figure 3 shows the pressure build up in the impeller chamber during priming. The objective is to avoid Point B, the maximum shut off head condition in order to minimize wear on the unit. This is done by opening the gates at Point A to allow a smooth transition to the pressure level C which is the operating pressure in the spiral case. Since level C varies with upper reservoir elevation the use of differential pressure insures that the maximum shut off head conditions can be avoided regardless of upper reservoir water elevation.

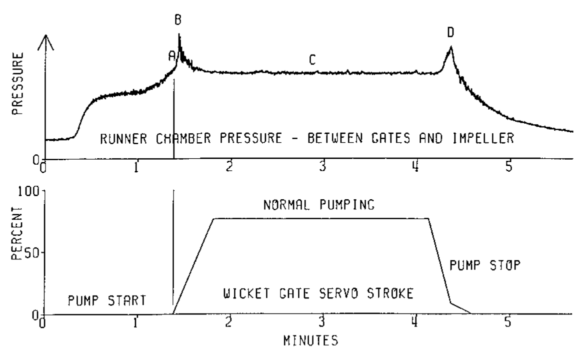


Figure 3 - Mode Changes

A short duration priming-pumping-shutting down cycle is shown in Figure 3. Such test information readily answers the question as to whether or not the priming cycle is optimized. On this plot there are two shut-off head spikes, points B and D. Point B occurs during priming and can be influenced. Point D occurs during shutdown and cannot be lessened without performing a pump power failure from about a 20 percent wicket gate opening which causes undue main breaker wear.

The idea is to select a wicket gate opening, Point A, so that Point B pressure is well below Point D in magnitude, without forcing high pressure water from the spiral case into the impeller. This plot indicates that the command for Point A should happen at a lower pressure. Since the

permissives for gate opening entail instrumentation, controls and computer dead time and verification time, the optimization of Point A becomes a trial process. Depending on the hydraulic design, the maximum pressure of shut-off-head, Point B can be more than 20 percent larger than the working pressure, Point C, so minimizing this daily transient pressure is a worthwhile goal.

Pump Testing at Zero Flow

To expedite the number of "primed" trials without using about 350 MW of system power input, a no flow test method was devised. This enabled control adjustments to be made and tested in a short time frame at Bad Creek.

For this test the wicket gates were manually and electrically prevented from opening. A visacorder chart recording was made of the pressure in the impeller chamber and the pick-up of the governor shut-down solenoid (which moved the distributor valve and opened the wicket gates) was detected by the event recorder (Figure 4). In this manner the response of the machine to changes in the analog or computer system were detected with respect to pressure build-up and wicket gate opening. The objective is to set control devices so that the event (opens wicket gates) occurs prior to achievement of shut-off-head pressure.

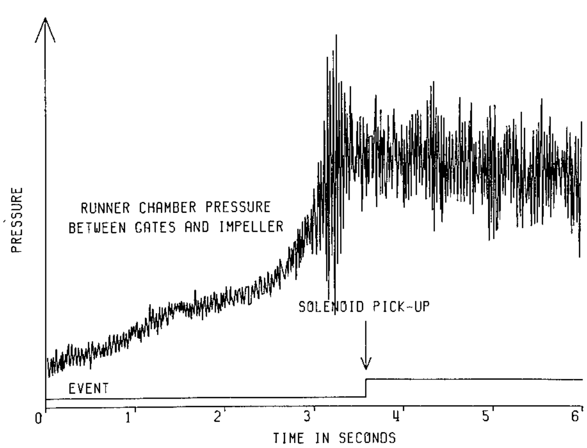


Figure 4 - Pump Start with Blocked Gates

This test procedure follows:

1. Secure wicket gates from opening.

2. Start unit in normal pumping mode which uses a pony motor.

3. Immediately after unit attains shut-off head pressure and the event marker trips - perform a pump to pump condenser mode changeover manually.

4. With unit rotating in compressed air and operating on the main motor, initiate venting of air and flooding for a normal pumping mode trial.

5. Repeat test as necessary until control or other checking is completed.

The advantages of this no flow test are in saving time in accelerating the unit to rated speed with the pony motor for each test trial and in requiring only 70 MW of input power since the wicket gates never open to establish flow. The system dispatcher appreciates the fact that only 70 MW rather than 350 MW are required which allows more flexibility in scheduling tests even in the day time.

Mechanical Balancing

The schedule for the mechanical balance of the rotating mass hinges on the availability of the driving medium: electrical power or water.

The first unit of a captive pumped storage station requires electrical power for balancing since no stored water exists for generate mode balancing and operation. Bad Creek employed the pony motor and main motor with the impeller rotating in air for the first unit to be commissioned. The succeeding units were done similarly despite the availability then of water in the upper reservoir for generator mode operation.

The engineering consensus favored balancing in air because hydraulic loads and vibrational vagaries were not present. In this manner the bearings and lubrication systems can be given operational checks up to rated speed without undue noise or trauma in the machine. Other considerations aside, balancing in air is easier on the nerves of the commissioning group.

At Mingtan, due to the manufacturing schedule of the static frequency starter, all mechanical balancing occurred by passing water to accelerate the rotating mass in the generate direction. No one can argue with the success of this method since it is always used for conventional hydraulic turbines. However it requires extreme vigilance since the bearing system and auxiliaries are unproven and are being subjected to significant centrifugal and hydraulic loads for the first time.

If the pump/turbine manufacturer could pick the ideal situation it would be to balance the unit in the air before operating with water in either pump or generate modes.

Conclusion

With each passing year the modern PC garners more field tasks since it not only stores data but can recall, analyze, plot, print, trend and be useful in numerous tasks with the only limitation being the imagination.

Historically, the method of rapidly opening the wicket gates to insure a fast breakaway for the establishment of thrust bearing oil film may not be necessary with the common usage of high lift pressure pumping systems for thrust bearings. Smoother transitions from the "at rest" to "synchronized" conditions may be obtained from a "slow-start" procedure which matches the acceleration of the rotational speed to the slower wicket gate opening position. Vibrational data on pump/turbines shows advantages for such slow-starts without trauma to any other unit components.

Optimization of the pump "primed" condition usually translates into saving wear on the machinery. Methods outlined in the text could be useful for anyone wishing to verify or better operation of a pump or pump/turbine. In some cases it was discovered that rehabilitated units have been operating at less than optimum conditions for decades with accumulated but unnecessary damage resulting. The Owner's could benefit from performing some of these simple tests.

The expedient of performing the mechanical balancing initially with water in the generate direction of rotation carries with it the necessity of repeating that balancing in the other modes of operation. The preferable method for initial balancing would be to rotate the unit with the pump starting equipment with the impeller rotating in air.

YARDS CREEK PUMP/TURBINE UPGRADE
PART 2: HYDRAULIC REDESIGN

S. Chacour[1] W. Colwill[2] J. Geuther[3] F. Harty[4]

ABSTRACT

The Yards Creek pump/turbines were commissioned in the mid-1960's. The hydraulic design of these units, while typical for their time is antiquated when compared with the latest pump/turbine technology. As part of an overall unit upgrade which included the improvement of several structural and operational problems (as presented in Part 1), a major design and model testing effort was undertaken to bring the Yards Creek pump/turbines up to modern, world class performance.

The station owners and their consulting engineer had prepared a specification for competitive bids, defining the project constraints and presenting a set of evaluation factors. The evaluation factors provided guidance to the pump-turbine designer for use in optimizing the combined value of the sometimes conflicting performance components such as efficiency, pumping discharge and generating capacity.

The design effort centered on a complete new design for the pump/turbine runner and redesigns of existing stayvanes and wicket gates. The designs were generated using a computerized design system, coupled with advanced flow analysis and experience gained from previous pump/turbine designs.

----

[1]President, American Hydro Corporation, York, PA 17402-3039.
[2]V.P. Marketing, American Hydro Corporation, York, PA 17402-3039.
[3]Senior Engineer - Generation, Jersey Central Power & Light Company, Yards Creek Station, Blairstown, NJ 07825.
[4]Principal Civil Engineer, Stone & Webster Engineering Corporation, Boston, MA 02107.

The model testing program was extensive and included two new runners, four sets of wicket gates, and four stayvane designs. Rapid numerical machining of the model components made it possible to improve successive designs based on current model results. Test results demonstrated the very significant improvements in pumping and turbining efficiencies achieved through custom redesign. Higher efficiency, plus a substantial turbine capacity increase, will provide a dramatic improvement in plant revenue.

This paper will describe in detail the pump/turbine hydraulic redesign, the model testing approach, and performance achievements for the various components. Comparisons with the existing unit performance will serve to demonstrate the improvements possible through upgrades of reversible pump/turbines.

BACKGROUND

The Yards Creek Pumped Storage Station was commissioned in 1965 as a nominal 330 MW station with three pump-turbine units. The original pump turbines were designed, built and installed by Baldwin-Lima-Hamilton. Gross head varies between 760 ft (231.7 m) and 687.5 ft (209.6 m). The station is operated by Jersey Central Power & Light Company (subsidiary of General Public Utilities) and is owned jointly by Jersey Central Power & Light Company and Public Service Electric & Gas Company. The station is located in northwestern New Jersey about four miles east of the Delaware Water Gap.

Construction of Yards Creek followed soon after the construction of Taum Sauk, making Yards Creek the second of the large capacity pumped storage stations developed in the United States. As such, it was one of the pioneers of the modern pumped storage era and, therefore, participated in the difficult experiences associated with this technological leap forward. The station had operated for over 25 years while experiencing and addressing difficulties with wicket gate vibration, runner vane cracking, excessive headcover deflection and stayvane cracking. Earlier remedial work had solved the problems of head cover deflection and runner vane cracking. In 1990, it was decided that the wicket gate and stayvane difficulties would be addressed as part of a hydraulic upgrade, which would also improve the generating and pumping performance of the station. The structural and mechanical upgrades of the pump-turbines are discussed in Reference 1. The present paper discusses the hydraulic aspects of this upgrade, which provide the most visible return for the investment in performing the work.

PROGRAM IMPLEMENTATION BY THE STATION OWNER

The implementation of the upgrade program was performed under the direct control of the JCP&L Station Engineer with assistance from PSE&G

and GPU specialists. The goals of the program were:

- reduced stresses in the stayring structure
- increased generating capacity
- increased pumping capacity
- increased efficiency in both modes of operation
- elimination of excessive wicket gate vibration
- reduced transient pressure rise
- increased service life

To supplement the owners' inhouse capability, Stone & Webster Engineering Corporation of Boston, Massachusetts was retained to provide consulting and assistance, including performance of strain gage measurements and fatigue and fracture mechanics analyses on existing components, preparation of the specification in a joint effort with the JCP&L Station Engineer, advice concerning prospective suppliers of the new pump-turbine components, technical evaluation of the bids, assistance in witnessing and evaluation of the hydraulic model tests, and advice and consulting throughout the program.

The model tests were conducted in 1991 followed by upgrade of the first Yards Creek unit scheduled for commercial operation during mid-1993.

SPECIFICATION

The specification provided for competitive bidding based on price and performance. The nature of pumped storage precludes the use of a simple weighted efficiency formula as is sometimes used with conventional turbines. With a pump-turbine the various performance components can have conflicting demands. For example, a design change which increases generating capacity may decrease pumping capacity or efficiency or both. The demands for pumped storage performance, therefore, require compromise. For this reason, it was necessary for the owners to develop a performance evaluation formula which would define a balance among the various competing demands for performance.

A calculation process was established for which each bidder would use its own guaranteed performance curves to calculate the following performance components:

- Rated capacity (average for two stated gross heads)
- Pumping energy input for a stated number of pumping hours (based on the power input and pumping discharge at three stated gross heads using the given head loss factors)
- Cost of pumping energy based on stated capital value per mwh.
- Round trip cycle efficiency based upon the gross head efficiencies at three different gross heads (using the given head loss factors).

- Value of energy generated according to a stated formula for operation over the guaranteed regulating range.
- Credit for the extent of the guaranteed regulating range.

The resulting calculation produced a quantity defined as the Total Relative Value (TRV), which was to be used initially for comparison of bids and later for comparison of the hydraulic model results with guaranteed performance.

Bidders were asked to quote on replacement of the pump-turbine runners and wicket gates (wicket gates for Unit 1 and 2 were to be reshaped), with the following options for the stayrings:

- Continued operation with the existing stayrings.
- Structural upgrade of the existing stayrings without modification of the stayvane geometry.
- Structural upgrade of the existing stayrings to include modification of the stayvane geometry.
- Complete replacement of the stayrings.

The successful bidder was American Hydro Corporation with a proposal that included a modification to the existing stayvane geometry to upgrade both the structural performance and the hydraulic performance.

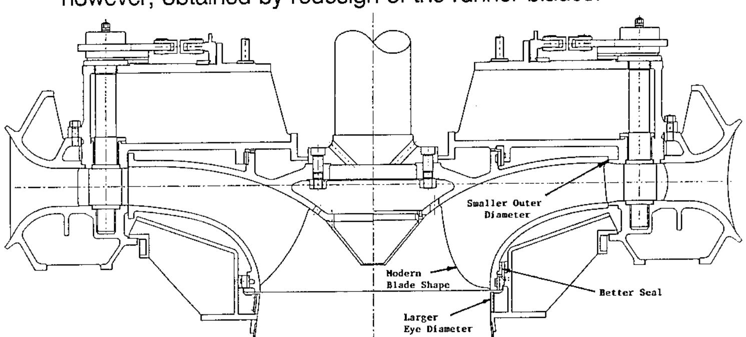
HYDRAULIC REDESIGN

A. Runner

The objective of the runner redesign was to increase efficiency in both cycles and to increase turbine power without increasing the pump input power. The existing Yards Creek runner was evaluated using both a statistical comparison with modern runners and the American Hydro Runner Design System (AHRDS)[2]. The following design characteristics were not consistent with modern design practice and were, therefore, improved in the redesign.

- The runner outer diameter was too large. This caused excessive disc friction, and decreased the efficiency.
- The runner passage height at the outer diameter was too small, which resulted in higher skin friction losses.
- The runner eye diameter was too small. The reduced area increased the through flow velocity and exacerbated the cavitation.
- The runner seal design could be improved to eliminate substantial leakage loss.
- The runner hydraulic passages were long and the runner blade leading edges (pumping and turbining) did not have the appropriate angles to match the incoming flow.

Figure 1 presents a comparison of the existing and upgraded runner designs. The combination of better seals and a smaller runner outer diameter was estimated to have increased pump/turbine efficiency by about 1.8 percent. The major change in runner performance was, however, obtained by redesign of the runner blades.



Yards Creek
Figure 1

For pump-turbines, the hydraulic design in the pump mode is compromised by the turbine operating conditions and vice-versa. The pump inlet blade angles must, for instance, match the incoming flow, to provide good cavitation performance, while these same angles will dictate the power achieved in turbine operation. The turbine inlet blade angles (at the outer diameter) should be kept more tangential to accommodate the lower turbine heads, while they must be sufficiently radial to allow operation at the maximum pumping head.

While the technology available for the original Yards Creek unit was not capable of carefully tailoring the blade design to best optimize pump/turbine performance, AHRDS provides the detailed flow analysis necessary to best contour the bucket shape to optimize the turbine performance while meeting all pump performance characteristics.

Working with the AHRDS system, the hydraulic designer first establishes the runner geometry. The interactive graphics provide complete flexibility in the design of the band, crown, and blade profiles. However, these shapes can only be optimized using a detailed knowledge of the flow field. The AHRDS system includes a three dimensional finite element flow analysis and a coupled boundary layer analysis to provide an extensive evaluation of the flow. For Yards Creek, many design shapes were tested using this analysis. Flow predictions in both pump and turbine operation were

used to optimize the final design. Particular attention was paid to optimizing pump cavitation performance and maximizing turbine high power efficiency.

Figure 2 presents comparisons of the Yards Creek original model performance (B-L-H data) and the performance of Yards Creek with a new runner but no other wheelcase changes. Here, and in subsequent figures the turbine performance is presented for 690 feet (210.3 m) of head which represents the middle of the turbine operating range. Based on the emphasis placed on turbine performance in calculating Total Relative Value, the new runner emphasized turbine performance and power while sacrificing some pump efficiency. The increase in turbine performance is quite remarkable, demonstrating the effectiveness of modern design techniques in maximizing turbine power while maintaining pump flow and power requirements.

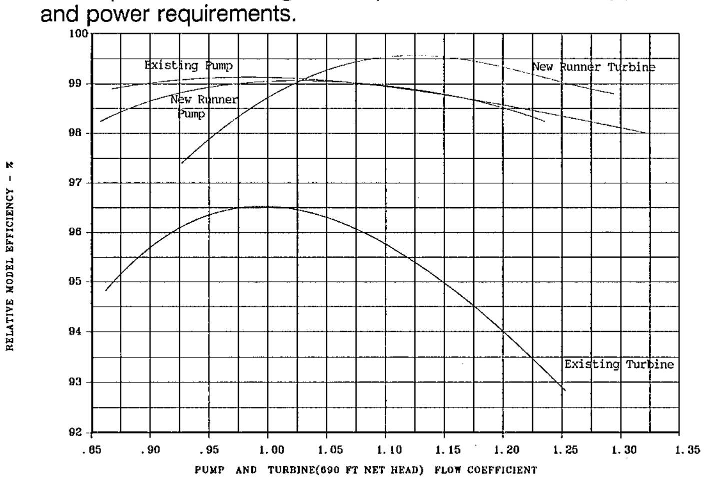


Figure 2 - Yards Creek Original BLH Model Data and New Runner

B. Wicket Gates

The existing spiral case and the runner each have specific requirements for the angular momentum (tangential velocity) at any given flow rate, though runner requirements will differ in pump and turbine operation. In the distributor, the stayvanes and wicket gates must be designed to match these requirements and provide a gradual transition of flow from spiral case to runner.

For turbine operation, the wicket gates must be designed to accept the flow from the stayvanes with minimal fluid incidence and deliver the correct angular momentum to the runner. In the pumping mode,

the wicket gate is adjusted to minimize the fluid incidence from the runner while delivering a flow which matches the stayvane inlet angle. Typically these pump and turbine requirements do not differ greatly. However, the optimum shape of the wicket gate will be specific to the stayvane shape.

For the original stayvane configuration at Yards Creek, the stayvane inner angle was rather large (from tangent). For the turbine at best efficiency operation, the fluid inlet angle to the runner was smaller than the stayvane's inner angle. To best match these requirements, the AHRDS design system was utilized to design and analyze various wicket gate designs. Final selection was based on both the potential flow and boundary layer analyses.

Both original and reshaped wicket gates were model tested. Figure 3 demonstrates the amount of performance improvement achieved through gate reshaping. The new shape could be machined out of the existing solid gates and while providing a substantial improvement in the dynamic characteristics as well. [1].

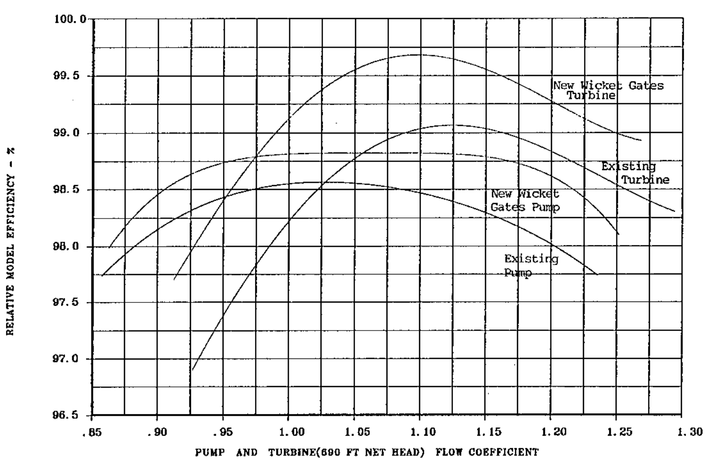


Figure 3 - Yards Creek Comparison of Existing and New Gates

Through the model test program it was found possible to extend the range of power output by increasing the wicket gate maximum angular rotation from 30° to 36°. Additionally, it was possible to shift the zero torque wicket gate angle below the synchronous-no-load position.

C. Stayvanes

In turbine operation, the spiral case presents a rather small flow angle (from tangent) at the stayvane outer diameter. The original stayvane had an angle which was much too large causing unnecessary fluid incidence losses. The modified vane provides a much better match over the majority of the vane (Figure 4).

The stayvane angle at the inner diameter should also be small to match runner requirements for either turbine or pump operation. The original Yards Creek vane, shown in Figure 4, was much too radial and cavitation of the existing vane in this location suggested a poor fluid incidence condition (pumping).

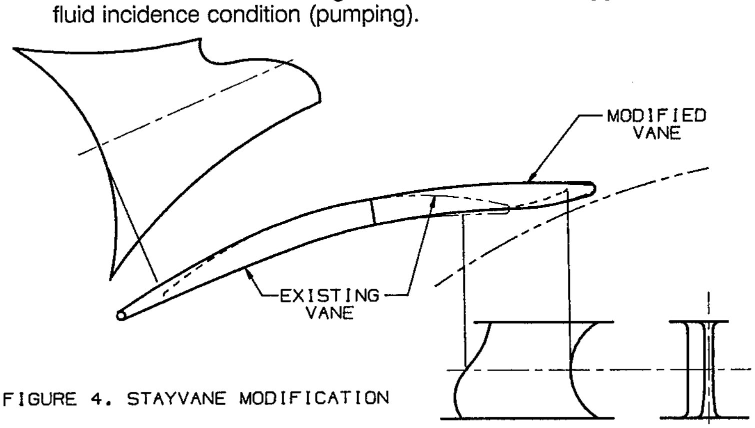


FIGURE 4. STAYVANE MODIFICATION

For the upgraded vane, structural integrity was paramount yet hydraulic considerations would have a significant effect on pump-turbine efficiency. The new vane insert (Figure 4) was developed jointly by mechanical and hydraulic engineers to provide sufficient strength at the inner diameter [1] and to have a small vane angle, thereby matching runner flow requirements. Test results shown in Figure 5 demonstrate the dramatic improvement in hydraulic performance resulting from stayvane reshaping. Reference 3 details the methods used to modify the prototype vanes.

FINAL TEST RESULTS

Figure 6 presents the final Yards Creek test results in comparison with the original Yard's Creek model data and with the contractual guarantees. As noted earlier, the value of turbine performance was worth much more to the owners than was the pump performance. The primary contractual guarantee was based on the calculated "Total Relative Value", and all design and test efforts were targeted at maximizing this result. It is evident that a substantial gain was achieved in the pump performance, although the

originally guaranteed values were not achieved. While the pump guarantees could be met with a different design of runner, stayvanes, and wicket gates, this "best pump" approach would have decreased the turbine performance and the TRV.

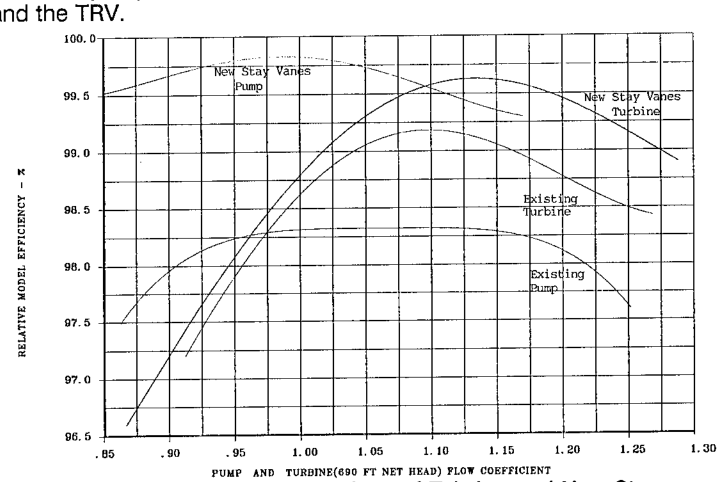


Figure 5 - Yards Creek Comparison of Existing and New Stayvanes

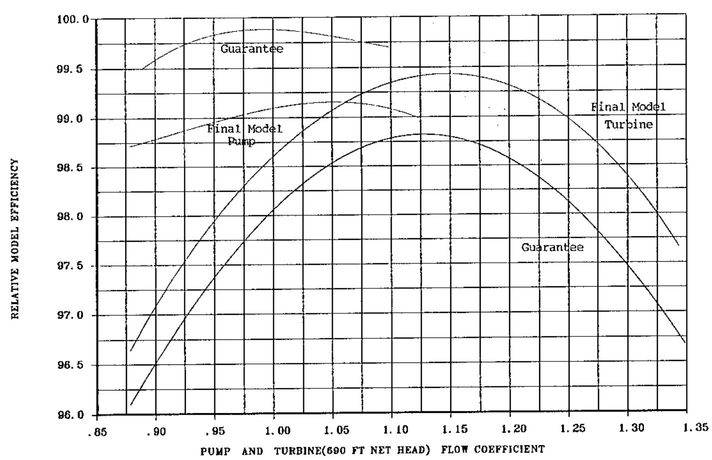


Figure 6 - Yards Creek Comparison with Guarantees

The upgraded turbine performance was clearly outstanding and represents the very substantial improvements possible when a pump-turbine is optimized for turbine performance rather than just being a "pump running backwards". While the peak efficiency of the turbine model hill curve was

increased, the major improvements were achieved by maintaining much higher efficiencies at heads below the peak and in increasing turbine output without increasing the pump power requirements. Because the turbine performance exceeded the guarantees, the Total Relative Value was substantially higher than the guaranteed value.

CONCLUSIONS

The Yards Creek hydraulic redesign has been most successful in raising the pump/turbine performance characteristics to efficiency and power levels close to those of modern new units. This result was achieved by:

- Establishment by the owners of a clear and concise method for evaluation of the pump turbine performance. This "Total Relative Value" provided a straight forward means of evaluating performance changes.
- Development by the engineer and the owners of a specification which included sufficient flexibility for the contractor to evaluate various upgrade possibilities.
- Complete hydraulic redesign of the runner, wicket gates, and stayvanes to match the existing waterpassages which is possible today using advanced fluid dynamic analysis and rapid computer interactive graphics.
- A detailed model test program allowed the testing and modification of all components. Rapid numerical machining of runners and wicket gates made it possible to evaluate test results and then redesign for better performance.

The entire project was set up and executed with substantial flexibility in what components could be changed and in the time spent on design and model testing. This flexibility was crucial in achieving the very high level of performance.

ACKNOWLEDGEMENTS

We wish to thank Mr. Jan Phillips and Mr. Richard Blashfield of GPU for their work in developing the plant performance evaluation.

REFERENCES

1. Degnan, J., Geuther, J., "Yards Creek Pump/Turbine Upgrade, Part 1 - Structural Redesign and Analysis", ASCE Waterpower Conference, 1993.
2. Chacour, S., Colwill, W., "Hydroturbine Runner Design to Maximize Power in Existing Wheelcases, ASME Paper No. 91-JPGC-Pwr-27, San Diego, CA, 1991.
3. Nolt, J., Corbit, R., Caploon, A., "Yards Creek Pump/Turbine Upgrade, Part 3 - Runner Manufacturing and Field Modifications", ASCE Waterpower Conference, 1993.

YARDS CREEK PUMP/TURBINE UPGRADE

Part 3 - Runner Manufacturing and Field Modifications

J. R. Nolt, Jr.[1] R. B. Corbit[2] A. Caploon[3]

ABSTRACT

In order to meet the Owner's high expectations and quality requirements relative to the supply of the upgraded runners for the Yard's Creek Pumped Storage Station, a unique approach to the runner fabrication was taken. Runners of this size, which have been fabricated in the past, have typically been constructed with cast components. These castings normally require a significant amount of upgrading by weld repair to meet most modern quality standards.

The techniques involved with using high quality stainless steel plate as the basic building block for the runner fabrication are described as well as the quality aspects involved. The problem of having to inspect the required quality level into the part, as is necessary with cast components, was eliminated.

Furthermore, the field modifications related to solving the fatigue cracking of the stayvanes are addressed. The material characteristics of the existing stayring placed notable importance on welding procedural development, along with a nondestructive examination program which could reliably detect defects larger than the maximum allowable flaw sizes.

RUNNER MANUFACTURING APPROACH

The decision to exclusively utilize high quality stainless steel plate as a basis for fabricating the runner components was made during the proposal stage of this project. The reasons for ruling out the use of castings are as follows.

----

[1]V.P. Operations, American Hydro Corporation, York, PA 17402-3039.
[2]Staff Consultant, GPU Nuclear Corporation, Middletown, PA 17057.
[3]Senior Engineer - Welding and Materials, Jersey Central Power & Light Co., Morristown, NJ 07960

1. Meeting the quality levels specified by the owner would be much easier if the basis for all of the fabrications were stainless steel plate. Ordinarily, with castings it is known that there will be a significant number of defects which require upgrading. The quality associated with castings is not inherent, as with plate, but instead, must be inspected into the piece. Furthermore, any volumetric inspection performed on a plate, with its uniform thickness, is much easier to carry out and therefore, much more reliable, as compared to castings with sections of varying thickness.

2. From a procurement standpoint, the plate would be readily available from several domestic sources, while the castings would only be available from a few sources outside the country.

3. Past experience indicated that manufacturing a runner from plate could be accomplished in a significantly shorter period of time than when castings were involved. Additionally, a much greater percentage of the work is under the control of the manufacturer which reduces the risk of late delivery. The chance of opening up casting defects during finish machining operations is also eliminated.

4. The techniques developed to effectively use plate in fabrications had been shown to be a more cost effective solution as compared to cast component utilization.

QUALITY REQUIREMENTS

The integrity of the runner design was verified through structural analysis. The maximum pumping and turbining modes of operation were first analyzed hydraulically to determine the pressure distribution on the runner's wetted surfaces. Based on these pressures and inertial body forces associated with 240 rpm operation in the gravitational field, deflections were found to be very small. The maximum reported tensile stress of 62 MPa exists on the vane-to-band fillet near the pump discharge. The analysis also assessed the runner's natural frequencies and respective mode shapes of vibration. No resonant amplification of the static stresses exists.

Fracture mechanics was used to determine minimum toughness requirements for the base material. Likewise a minimum charpy impact strength of 41J at 0°C was specified for the weld materials along with a minimum ferrite number of five.

A key aspect of the runner manufacturing process was the implementation of the contract quality requirements into a working document. This quality Control Plan, which was approved by the Owner, established notification points, outlined the procurement requirements, and stipulated all of the nondestructive testing requirements.

Since an extensive amount of welding and nondestructive examination were required by this project, welding controls and examination requirements were very important. Welding procedures and welders/operators were qualified to Section IX of the ASME Boiler and Pressure Vessel Code. Those individuals performing the nondestructive testing were required to meet the qualification requirements of ASNT-TC-1A. Section V of the ASME Code provided the guidelines for the nondestructive examination procedures. The fracture mechanics analyses were used to determine the minimum acceptable flaw sizes.

RUNNER FABRICATION

Initial manufacturing operations were performed concurrently on the crown, band, and vanes. Each of these components was fabricated with plate material conforming to ASTM A240 UNS S30403. The illustration in Figure 1 shows the locations of the welds used to join the various component plate members. A description of the fabrication of each component follows.

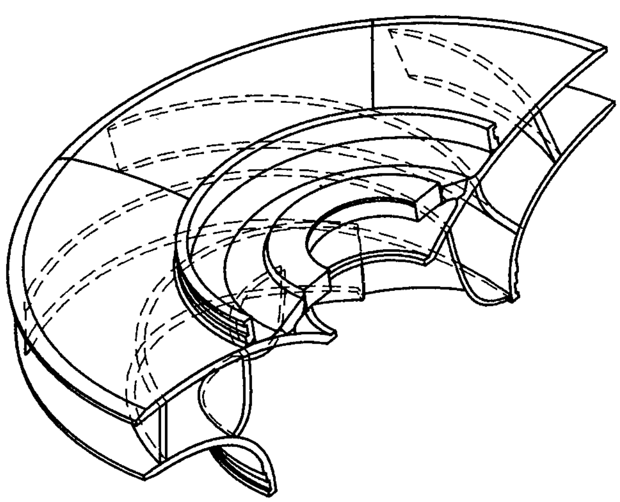


Figure 1 - Sectionalized Runner Illustrating Weld Joint Locations

Band - The 5.4 m runner band was fabricated from six 90 mm thick plates. Forming dies were fabricated and machined so that each of these plate segments could be hot formed to the precise shape. Figure 2 shows the formed segments which were machined to provide the required weld geometry at the butt joints.

Joining of the segments was accomplished with the flux-cored-arc welding process on a welding positioner, which made welding possible in the flat position for improved weld deposition rates and quality as shown in Figure 3. Following the welding operation, the water passage surface of the band was CNC machined to the final design contour.

Crown - As with the band, hot die forming of 90 mm thick plate provided six pie-shaped segments which were joined by flux-cored-arc welding. Meanwhile a subassembly of the bolting flange and the transition piece was fabricated. Heavy 203 mm thick plate was plasma cut for the flange. The transition piece was plasma cut and cold formed into a cone from 146 mm thick plate.

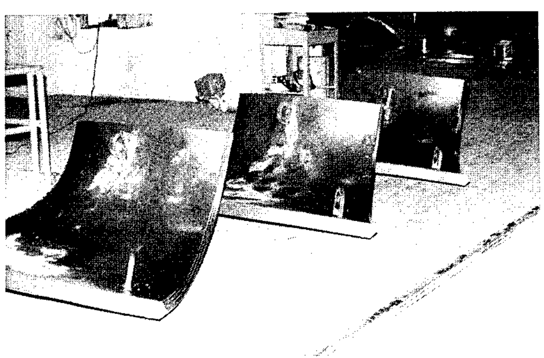


Figure 2 - Formed Band Segments With Machined Weld Geometry

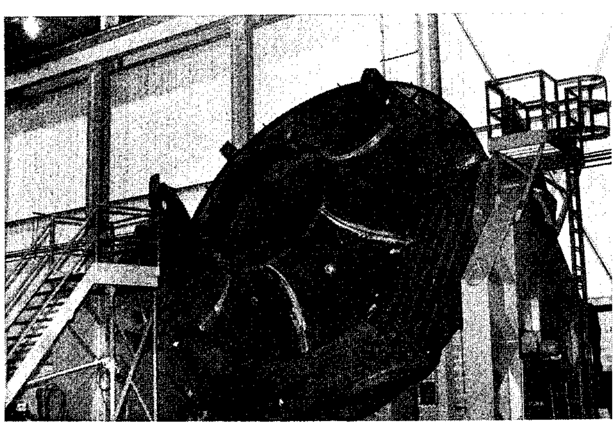


Figure 3 - Assembled Band on Welding Positioner

A cylindrical wearing ring holder was formed from 114 mm thick plate and the joining of this piece and the flange subassembly to the hot-formed section was accomplished with submerged arc machine welding. The top side of the crown showing the flange subassembly can be seen in Figure 4.

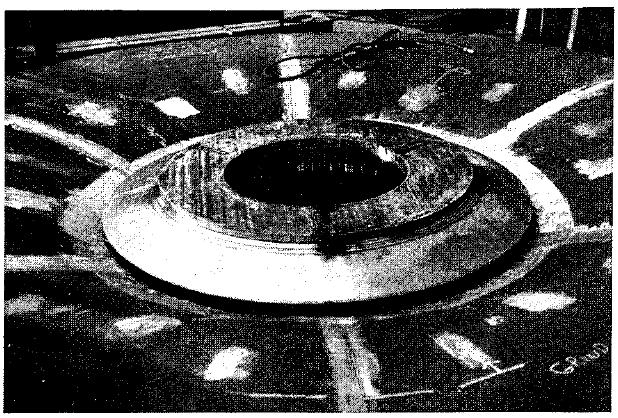


Figure 4 - Runner Crown Bolting Flange Subassembly

The upper portion of the 5.4 m diameter crown was rough machined and the water passage side finish machined on a CNC vertical boring mill. Boring of the pressure equalization holes was subsequently performed on the large 5-axis machining center pictured in Figure 5.

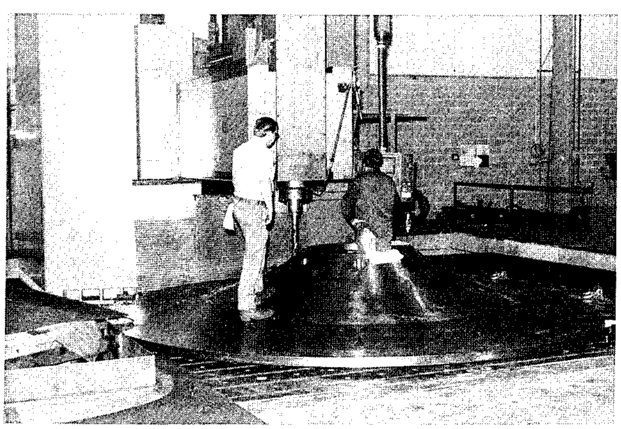


Figure 5
Drilling Pressure Equalization Holes
In Runner Crown On 5-axis Milling Machine

Vanes - The techniques developed to manufacture the vanes rely heavily on computer software which has fully automated the process. Using the computer designed vane, the program develops it into a flat shape, which is then automatically nested on a plate to minimize scrap loss. After the nesting sequence, a CNC plasma cutting program is automatically generated and sent via a direct link to the plasma machine, which cuts the plate.

Machining of these flat shapes is performed to obtain the designed thickness distribution. Again the numerical control files are automatically generated and transmitted to one of the machining centers.

Hot forming of the vanes between CNC machined dies is followed by machining of the pump entrance and pump discharge edges as well as the weld geometry at the intersection with the crown. Past experience has shown that vanes, which are manufactured in this manner, are not only superior to cast vanes from a structural integrity standpoint, but also excel from the aspect of dimensional accuracy. The latter point holds true even when cast blades are totally machined.

Runner Assembly - The seven vanes were attached to the crown using assembly fixtures which accurately provided the vane locations. Figure 6 shows this assembly on a welding positioner. Flux-cored-arc welding with filler material conforming to ASME SFA5.22 Class E308LT-1 was employed to make these complete penetration joints. Weld and heat-affected zone toughness properties measured from the procedure qualification involving some of the actual runner material are displayed in Table 1. The values greatly exceed the minimum design requirement of 41J at 0°C.

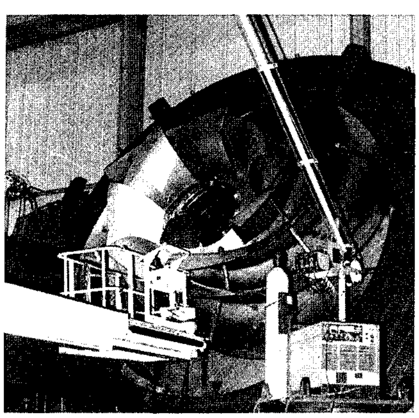


Figure 6 - Crown - Vane Assembly On Welding Positioner

TABLE 1 - Runner Toughness Values Measured at 0°C

Location	CVN, J
Weld Metal	104
HAZ	130
Base Metal	>271

After premachining the band edges of the vanes and assembling the band, welding was performed in a similar manner. Following both dye penetrant and ultrasonic examinations of 100 percent of the weld joints, the runner was finish machined and dynamically balanced in the manner shown in Figure 7. Experience has shown that the amount of balance correction is significantly less than with runners fabricated from castings.

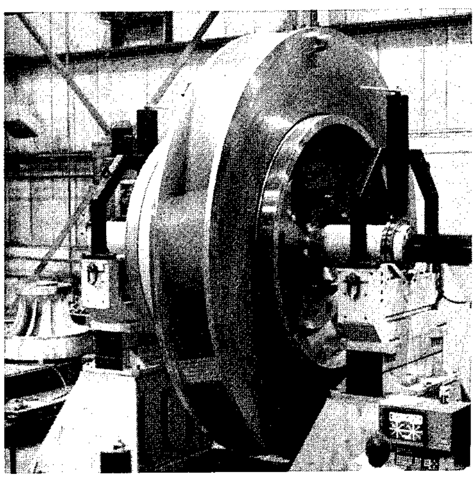


Figure 7 - Dynamic Balancing of Runner

STAYVANE MODIFICATIONS

The main purpose in modifying the stayvanes was to eliminate the fatigue cracking problems which have plagued the plant in recent years [1]. The modification was, however, designed to improve the hydraulic performance of the machine as well [2].

The solutions involved welding a uniquely designed stayvane insert, thus replacing the existing pump entrance edge of each stayvane, and altering the shape at the pump discharge edge through the removal of existing material.

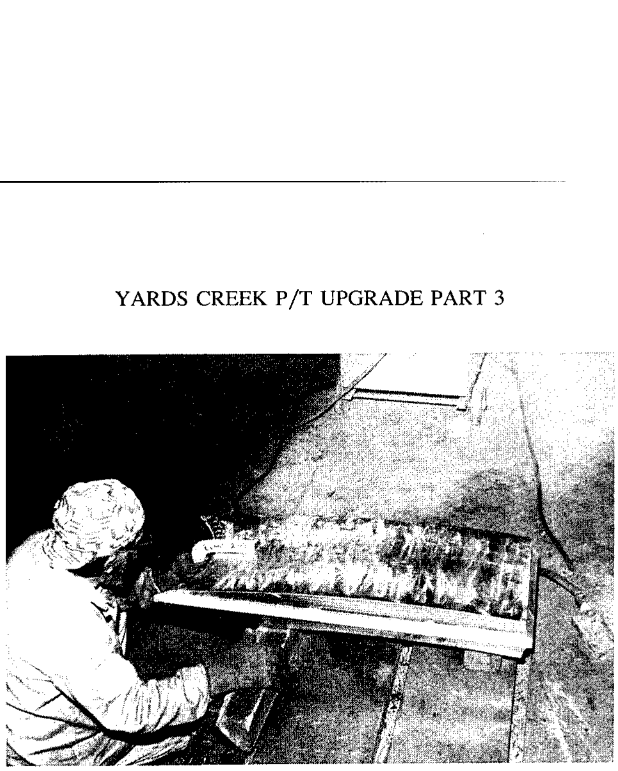


Figure 8 - Stayvane Insert

The material chosen for the insert was ASTM A240 UNS S30403 which has extremely good toughness and excellent corrosion resistance. Its low yield strength and high ductility would allow it to locally yield to reduce peak stresses in the stayring structure. The inserts were machined and ground to the shape modeled in the finite element stress analysis. A picture of one of the inserts with machined weld joint preparations is shown in Figure 8.

The major challenge associated with this repair was welding the stayring casting which was manufactured to ASTM A148 Grade 90-60. The results of several chemical analyses of the Unit 2 stayring are displayed in Table 2. Prior testing of this material indicated that its toughness was extremely poor (in the order of 7J at 0°C)[2] [3].

TABLE 2
Unit 2 Stayring Chemical Composition (Analyses of Several Samples)

C = 0.31 - 0.32	Mn = 1.53 - 1.57
Si = 0.29 - 0.30	S = 0.13 - 0.15
P = 0.014	Ni = 0.002
Cr = 0.10 - 0.11	Mo = 0.41

Relying on the Owner's past experience when repair welding this material, a procedure was jointly developed which would allow welding the inserts without the need for a postweld heat treatment, and minimizing the exposure of the welders to uncomfortably high preheat conditions.

A 560 mm long section of each existing vane was removed by oxy-acetylene cutting. The gas cut surfaces were ground and inspected by magnetic and ultrasonic examinations. Any indications were removed and repaired. A butter layer using electrodes conforming to SFA5.4 Class E309L-16 was deposited on base material in the areas which would ultimately become part of the weld joint. A 177° C preheat was applied and maintained during the buttering operation using thermocouple controlled electric resistance preheat elements. Surface inspection was performed on the adjacent base material at regular intervals while the butter layer was being applied.

Following its application, grinding of the butter layer was carried out, followed by a dye penetrant examination of this surface. Using a carefully controlled welding sequence, the stayvane inserts were welded into place with electrodes meeting the SFA5.9 Class E308L-16 specification at basically ambient temperature. In addition, selective peening was applied to control distortion. An insert with the welding nearly completed can be seen in Figure 9. After final grinding of the fillets, both dye penetrant and ultrasonic tests were performed on the welds. Toughness values anticipated for the various regions of the weldment are listed in Table 3. These data were measured from the procedure qualification which incorporated actual stayvane material from Unit 2.

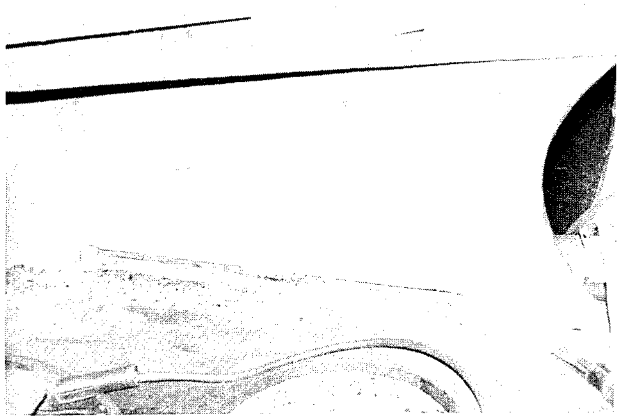


Figure 9 - Insert With Welding Nearly Completed

TABLE 3
Stayring Insert Welding - Toughness Properties Measured at 0°C

Location	CVN, J
Weld Metal	111
A148 HAZ	69
304L HAZ	145
Butter Layer	71
A148 Box Metal	7

CONCLUSIONS

The techniques applied to manufacturing the replacement runners for Yards Creek have resulted in the supply of runners with a higher level of structural integrity and improved dimensional accuracy, than has been typical in the past.

The modifications involved with addressing the stayvane cracking problem were successfully implemented. The welding procedural development applied to joining the stayvane inserts to the existing stayring material, proved to be effective.

REFERENCES

1. "Final Design and Analyses of Yards Creek Stay Ring, Wicket Gates and Impeller", American Hydro Corporation, April 7, 1992.

2. Chacour, S., Colwill, B., Geuther, J., Harty, F., "Yards Creek Pump/Turbine Upgrade Part 2 - Hydraulic Redesign", ASCE Waterpower Conference, 1993.

3. "Yards Creek Modified Unit #1 Stay Ring/Vane Failure Analysis", Allis Chalmers Corporation, May 16, 1985.

Central Valley Project Energy Efficiency Study

Martin Bauer[1] and Guy Nelson[2]

Introduction

The Central Valley Project (CVP) was developed to utilize the water resources of the Central Valley of California. Surplus water from the Sacramento River and its tributaries in the northern part of the valley is stored in a series of reservoirs. Canals transport the water to other areas of the San Joaquin Valley.

The multipurpose project provides irrigation, municipal and industrial water, hydroelectric power, and flood control. The project also provides fish and wildlife enhancement, recreational opportunities, improved navigability of the Sacramento River, and control of saltwater intrusion into the Delta area.

The CVP is a major user and generator of power. An efficiency savings of 1% of the total project's daily power use is enough to supply power to 7000 homes for the day. Energy efficiency is a major concern of the CVP. Therefore, the Western Area Power Administration (Western) and the Bureau of Reclamation (Reclamation) formed a study team in 1991 to explore energy efficiency improvement opportunities within the CVP.

In the initial phase of the study, the project team identified all project use loads. Project use loads include those loads which are critical to providing the basic service of delivering water, are authorized by Congress, or are authorized projects which Reclamation maintains.

---

[1]Electrical Engineer, Bureau of Reclamation, 2800 Cottage Way, Sacramento, CA 95825.

[2]Energy Services Manager, Western Area Power Administration, 1825 Bell Street, Suite 105, Sacramento, CA, 95825.

The team limited the scope to six facilities which consume the bulk of the CVP project use power (85%). This was necessary to provide sufficient time for on-site evaluation, data analysis, and preparation of the report by the prescribed completion date. Table 1 lists the sites which were considered for pumping efficiency improvements.

The team determined recommended power operations improvements from efficiency projects which have been identified as economically sound. Improvements that would result in operationsl benefits, but not economic benefits were not listed. The team did not perform site visits of power facilities; study activities did not require field investigation, and the time available was limited.

Conclusion

The team found that many efficiency improvements already have been implemented by Reclamation and the California Department of Water Resources. Increased efficiency of pumping plant operation has been a priority since the early 1980s. Efficiency improvements have been made in the course of replacing equipment due to deterioration and as part of the efficiency improvement programs. Table 2 illustrates the projects and their returns identified to improve the efficiency of the CVP.

Pumping Facility Improvements:

Each site was evaluated to identify potential improvements.

- In most cases the efficiency of the impellers were up to current standards. A program of replacement under way for those impellers suffering from deterioration.
- Analysis of excitation system replacement at all sites indicated that the costs of the replacement would not be justified when compared solely against the economics from efficiency gain.
- Automatic debris removal systems did not provide sufficient energy gain to justify installation.
- Adjustable speed drives or variable pitch impellers also did not provide sufficient economic return to warrant installation.
- At one site, modifying the speed of the several units eliminates a poor efficiency characteristic and has a rate of return which warrants the project.

- Retrofitting with newer, higher efficiency transformers, or reducing pumping head losses by modifying the discharge systems did not appear to provide sufficient energy gain to warrant consideration.

Power Operation Improvements:

The list of recommended improvements covers only projects which have been previously analyzed and found to be economically sound. Some improvements at power generation stations not included have been found to provide operational benefits to the power system with economics difficult to quantify.

Following is a list of conclusions specific to each site:

1. Replace the existing impellers at Tracy with higher efficiency and capacity impellers. The higher capacity impellers will provide enough additional capacity that only five of the six units will be needed for the current maximum water delivery. In an average year approximately 19,000 MWh of energy will be saved.

2. Replace three of the six variable vane impellers at O'Neill with higher-efficiency fixed-vane impellers. This would still provide sufficient flexibility of operation along with an approximate average of 1,600 MWh of energy saved each year.

3. Develop a control system for the O'Neill Pumping Plant which would allow the optimization of the variable vane unit characteristics. This would result in further savings with or without impeller replacement with fixed-vane impellers.

4. Modify two units at Gianelli (formerly San Luis) pump generating plant to operate at higher speeds. This would significantly reduce the losses when pumping at high heads. Two units have already been modified and are operating as predicted.

5. Increase the height of the Keswick reservoir. This would allow an increase in power production at Keswick of approximately 1,000 MWh per year in an average year.

6. Install a new water line to the Nimbus Fish Hatchery. This would allow the Nimbus Lake to provide greater regulation of Folsom Powerplant releases. The energy generated by Folsom would offset peaking purchases and would result in an average annual savings of $450,000.

7. Uprate the Shasta Units. Units 3, 4, and 5 would make use of water which is normally released through the spillway at the top of the dam during high-flow periods. The energy savings

averages approximately 24,000 MWh per year, based on historic operation.

8. Install a device which allows the water, that bypasses the Shasta powerhouse, to be used to optimization of water temperature in the Sacramento River for fish and to generate electric power. This would allow an increase in power production at Shasta of approximately 285,000 MWh per year in an average year.

Table 1

Summary of Central Valley Project Electrical Use

Project Load	Installed Capacity (kW)	Avg. Yr. Energy Used (MWh	Average Yr % of Total CVP
Tracy Pumping Plant	103,228	650,000	40.36%
H.O. Banks (Delta) Plant	261,515	113,000	7.02%
O'Neill Pumping Plant	26,856	87,000	5.40%
Gianelli Pumping Plant	202,912	270,000	16.76%
Dos Amigos Pumping Plant	179,040	200,000	12.42%
Pacheco Pumping Plant	17,904	31,000	1.92%
All Other Loads		259,509	16.11%
Total Project Loads		1,610,509	100%

Table 2

Summary of Recommendations

Improvements	Total Capital Cost ($ K)	Power Savings (GWh)	Internal Rate of Return (%)	(30 yr.) Effective Mill Rate
1. Tracy Impeller Replacement	3,000	19	30	15
2. O'Neill Impeller Replacement	900	2	<1	51
3. Keswick Splashboards	300	1	15	28
4. Nimbus Fish Hatchery Line Replacement	1,300	Shift	70	1
5. Shasta Powerplant Generator Uprate	2,000	24	57	8
6. Shasta Dam Temperature Control Device	45,000	285	25	15

STRATEGIC, OPERATING AND ECONOMIC BENEFITS OF MODULAR PUMPED STORAGE

Rick S. Koebbe*

ABSTRACT

Hydroelectric pumped storage has been proven as the most efficient and cost-effective technology available for energy storage. It is used by utilities to provide peak and intermediate power and to optimize overall system performance. Because of increased environmental and regulatory constraints, limited acceptable sites, long permitting and construction schedules, and huge financial commitments, large conventional pumped storage plants, typically 1000 MW or more, are now virtually impossible to build on existing waterways. As an alternative, Peak Power Corporation created Modular Pumped Storage™, with project sizes ranging from 50 MW to 200 MW, to overcome the difficulties associated with large conventional projects. The modular approach involves standardizing the elements of a pumped storage plant by utilizing specialized siting techniques and optimizing design, equipment, and construction. Compared with conventional pumped storage, the modular design reduces cost and expedites schedule, reduces environmental concerns and permitting obstacles, and expands applications of energy storage on a utility's system. Modular Pumped Storage has been designed to compete in today's power market, not only with conventional pumped storage, but directly with gas turbine plants—in cost, schedule, location, and project size.

---

*Vice President, Peak Power Corporation
10 Lombard Street, Suite 410, San Francisco, California 94111
Telephone 415-362-0622

INTRODUCTION

Hydroelectric pumped storage, or pumped storage, is widely recognized as the most mature and efficient energy storage technology available. There are more than 180 pumped storage plants in operation worldwide with a total installed capacity exceeding 70,000 MW. Over the past 50 years, more than 19,000 MW of pumped storage capacity in 37 plants has been installed in the United States, equating to roughly 3% of the total installed generating capacity. There are also many other projects in the planning stage.

The growth of pumped storage capacity in the 1970s and 1980s reflected the advent of high capacity baseload generating plants, a widening differential between the cost of on-peak and off-peak energy, and the increasingly uncertain growth of peak demand. Although they were originally built to primarily complement nuclear plants or to solely provide peaking capacity, most pumped storage plants are now economically dispatched to manage overall system performance. By providing load leveling and non-thermal load following (thereby reducing thermal plant cycling and minimizing loading inefficiencies), operating and spinning reserve, voltage and power factor correction (VAR support), frequency regulation, and other operating benefits, hydroelectric pumped storage plants have improved the overall efficiency (or heat rates) of their host systems while also reducing thermal plant operation and maintenance costs.

Conventional hydroelectric pumped storage plants, however, are usually cost effective only at large capacities; most of the plants built or planned over the past 20 years have been 1,000 MW or more. These projects have been impeded by long lead times (permitting typically requires six years and design and construction an additional five years [source: Electric Power Research Institute]), large financial commitments (typically greater than $1 billion), few acceptable sites, and poor public acceptance due to environmental concerns. The shear size of these plants has effectively denied most utilities the advantages of energy storage.

Modular Pumped Storage™ has been developed and designed by Peak Power Corporation to compete in today's power market, not only with conventional pumped storage, but directly with gas turbine plants—in cost, schedule, location, and project size. Modular Pumped Storage is designed to be economically competitive with simple-cycle gas peaking units; to have expedited project schedules, typically less than five years;

to be sited in the immediate vicinity of load centers; and to range generally from 50 MW to 200 MW in project size, providing critical system control and unlimited dispatchability.

MODULAR DESIGN

Modular Pumped Storage (MPS) is designed to overcome the siting and project development difficulties which inhibit installation of conventional hydroelectric pumped storage plants by:

- Being located off-stream and avoiding impacts to rivers or lakes;
- Minimizing environmental impact at the off-stream sites;
- Requiring relatively small land areas; and
- Achieving economies in siting, equipment, and construction that enable MPS projects to be built at less cost ($/kW) than large conventional pumped storage plants.

To achieve the objective of competing, not only with conventional pumped storage, but with thermal peaking plants, the MPS design incorporates two relatively small artificial reservoirs in a closed hydrologic system. The reservoirs require no access to rivers or lakes and have no impact on water quality or aquatic life; groundwater is typically utilized as the water source. Locating projects away from rivers substantially reduces environmental impacts, provides flexibility to site projects, simplifies and expedites permitting, and contributes to cost and schedule certainty.

The MPS plant and equipment designs are engineered to be modular and standardized, within predefined head ranges. This also simplifies engineering, expedites schedule, and creates cost savings. Small modular projects allow short construction schedules, and enable even small and mid-size utilities to take advantage of a technology previously available only to large utilities.

Siting a MPS project is driven largely by the necessity to be cost-competitive with thermal peaking projects. To achieve least cost, the best sites for Modular Pumped Storage projects typically include a hydraulic head of more than 800 feet and a horizontal distance between reservoirs of less than 8000 feet. High heads keep equipment and construction costs low and required water storage volumes to a minimum. Prospective MPS sites are evaluated with a focus on utilizing existing topographical and geological features to the greatest

extent possible to substantially reduce the civil costs associated with dam/reservoir construction. Reservoirs are typically sized for a daily generation-storage cycle with 2000 MWh of storage.

Proximity to load centers and to existing transmission systems are other important criteria for siting a MPS project. MPS sites are located as close to major load centers as possible, thereby reducing transmission costs and allowing the utility maximum use of the strategic and operating benefits.

Competitive Cost and Schedule

Modular projects are relatively small compared with conventional pumped storage projects. The traditional wisdom, however, asserts that "small" is not economically feasible or cost competitive. But based upon the modular design criteria, a 100 MW or 200 MW pumped storage project typically has a capital cost of roughly $600 per kilowatt. This includes all direct costs such as engineering and design, land, equipment, construction, and transmission, but excludes indirect costs such as interest during construction. By comparison, a conventional 1000 MW pumped storage project typically costs over $1000 per kilowatt. The difference in cost between small modular projects and large conventional projects is shown in Figure 1. A small modular project has lower costs primarily due to the standardized modular design which includes economies in siting, equipment, and construction.

Modular Pumped Storage provides economic, operational, and strategic benefits and values that gas-fired peaking units generally do not. A description of these benefits and values is provided later. Studies by utilities have proven that the levelized operating cost of a MPS project is economically competitive with simple-cycle gas peaking units, especially when consideration of the strategic, operating, and economic benefits of energy storage are fully evaluated.

Schedules of large conventional pumped storage projects typically range from eight to sixteen years. The lengthy schedules create huge financial risk. And because the project's on-line date is over a decade away, the new capacity cannot be accurately correlated with the utility's load forecast. A small Modular Pumped Storage project, by contrast, can be built in roughly four to five years, as shown in Figure 2.

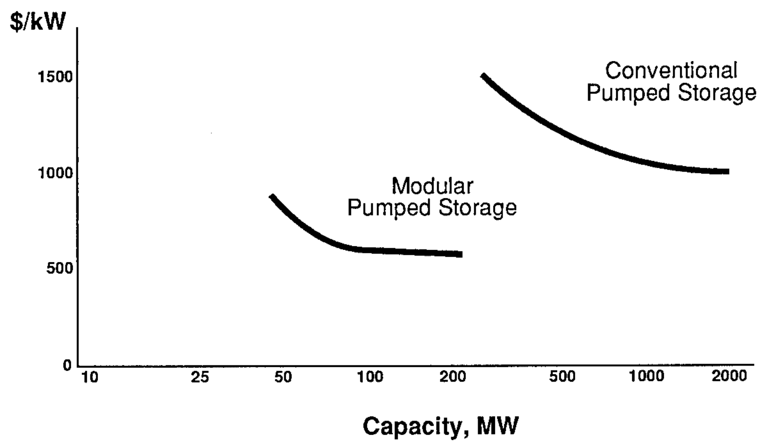


Figure 1. Cost Comparison of Modular v. Conventional Pumped Storage

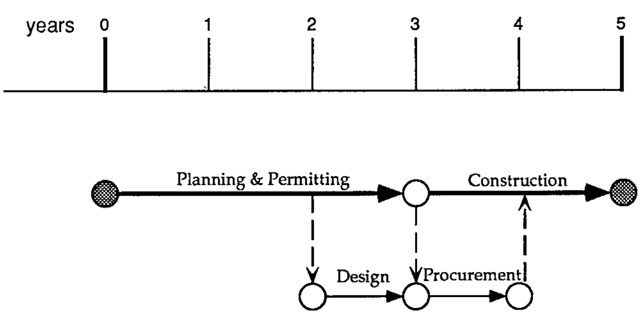


Figure 2. Modular Project Schedule

ECONOMIC BENEFITS

A pumped storage project's dispatchable pumping load will enable the most efficient plants on a utility system to operate more hours every day. The base-load energy stored during light load hours can then be used to displace operation of the most expensive plants during peak load hours. Pumped storage also allows unit commitment to be optimized, improving incremental heat rates and reducing production costs. The economic benefits of pumped storage are derived primarily from:

- Economic energy transfer, achieved by using the least-cost energy available on the system (stored via pumping) to displace higher-cost energy;

- Reduced cycling of base-load thermal plants. Keeping marginal thermal units baseloaded during off-peak hours increases the incremental efficiency of these units significantly. Plus, less cycling reduces operation and maintenance costs and extends equipment life;

- Meeting regulating margin and displacing operating and spinning reserves with non-thermal pumped storage. Pumped storage's quick-start capability qualifies as operating reserve (in the stand-still mode) and spinning reserve, saving fuel cost and reducing or eliminating emissions from thermal plants. This also reduces thermal plant startups, extends equipment life, and reduces maintenance costs;

- Efficiency gains from displacing less efficient thermal plants and reduced unit starts; and

- Firming up economy energy and/or non-firm intermittent resources such as wind.

To illustrate the value of these economic benefits, assume that a 200 MW Modular Pumped Storage project will be integrated with a moderately sized utility with a peak load of roughly 3000 MW, with excess coal-fired baseload capacity, and a growing peak demand.

Collectively, these economic benefits could reduce this utility's production cost by $10 million to $20 million per year, or $50/kW to $100/kW of equivalent capital cost. For example, a 200 MW Modular Pumped Storage project could increase the capacity factor of this utility's 800 MW coal plant from 68% to 82%. The resulting increase in thermal efficiency can save approximately 250,000 tons of coal or about $5 million per year. Utilizing the efficient stored energy from the coal plant instead of the less efficient and more expensive "cycled" output could reduce production cost by an additional $10 million per year.

DYNAMIC OPERATING BENEFITS

A pumped storage project's operating and system control capabilities will enhance the management of a utility's system. The value of these capabilities are a function of each utility's operating characteristics. Based upon the assumptions provided earlier, the dynamic operating benefits of pumped storage to this utility can be estimated using an EPRI framework as follows:

Dynamic Operating Benefits (estimated)
200 MW Modular Pumped Storage Project

Operational Req't	*Value $/kW*	*Basis*
Frequency Regulation	5	Regulating service typically met with thermal resources
Load Following	25	Least cost source; frees up 200 MW dedicated capacity
Operating Reserve	30	Lowest-cost way to meet required reserve
Voltage Regulation	5	Power factor control; transmission VAR support
Ramping	10	Some thermal units now cycled
Black Start	5	Value equal to dedicated gas-fired peaking unit
Total	$80/kW	

To illustrate the economic benefits with this utility, assuming a fixed charge rate of 10.3% (source: EPRI), this $80/kW equivalent capital cost translates into a benefit of $8.24/kW-year. This means that the 200 MW Modular Pumped Storage project should receive an annual credit of $1,650,000 per year in evaluating its impact on the total cost of operating the utility's system. Depending upon the unique characteristics of each utility's system, there also may be additional operating benefits not identified in the table above.

STRATEGIC BENEFITS

In comparison to other resource alternatives, Modular Pumped Storage also provides strategic benefits to a utility. These include:

- Hedge against rising fuel prices, derived from the ability to use the lowest cost energy available to displace the highest cost energy;
- Reduced net emissions, reflecting improved thermal plant efficiencies and less fuel being burned to meet the load;
- Replaces hydroelectric power used as an intermediate and peaking resource in "dry" water years;
- Hedge against changing economy energy market conditions;
- Least environmental impact peaking technology;
- Ability to firm up and accept larger amounts of renewable non-firm intermittent resources, such as wind;
- Ability to provide full dispatchability and accept larger amounts of renewable baseload resources, such as geothermal;
- Reduced transmission constraints and increased transmission efficiency; and
- Avoided or deferred transmission construction. The siting flexibility of Modular Pumped Storage allows project locations to be selected specifically to achieve these kinds of transmission-related benefits.

The value of these benefits, which are a function of the characteristics of each utility's system, can be expected to be at least $70/kW of equivalent capital cost.

Production cost modeling can quantify the pumped storage project's economic benefits—that is, the savings produced by adding pumped storage to the resource plan compared to adding an equivalent amount

of thermal peaking capacity. Additional studies to identify the improvements in reliability, transmission system efficiency, air quality, and other factors may be necessary to fully quantify the pumped storage project's operating and strategic benefits.

Based upon utility experience with pumped storage plants, the combined strategic, operational, and economic benefits of a Modular Pumped Storage project can be estimated to be in the range of $200/kW of equivalent capital cost (i.e., $50/kW economic benefits + $80/kW operating benefits + $70/kW strategic benefits). This estimated value would reduce the effective direct cost of a Modular Pumped Storage project from $600/kW to $400/kW; the effective installed cost would be reduced from $750/kW to $550/kW.

CONCLUSION

The Modular Pumped Storage design has been refined and optimized through the contributions of numerous engineering firms, equipment manufacturers, and construction firms, with experience on many of the major hydroelectric and pumped storage projects built over the past 20 years. There are no new technological risks.

Unlike conventional projects, Modular Pumped Storage projects can reduce environmental impacts, expedite schedules, and provide competitive costs at relatively small plant sizes (50 MW to 200 MW), with overall reduced risks. The modular design utilizes a closed hydrologic system, located off-stream and independent of existing rivers or lakes, utilizing two artificial reservoirs, and incorporating standardized siting, equipment, and construction criteria. Modular Pumped Storage has been designed to compete favorably in today's power market against gas-fired projects as an alternative peaking resource, while providing unique economic, operating, and strategic benefits to the utility. The modular project size both expands applications of pumped storage on a utility's system and allows small and mid-size utilities to take advantage of a proven and efficient energy storage technology previously available only to large utilities.

Peak Power Corporation is currently developing over ten Modular Pumped Storage projects in nine states with public and investor-owned utilities, irrigation districts, electric co-ops, and independent power producers.

The author graduated from The University of Michigan, Mechanical Engineering program.

REFERENCES

ASCE, "Civil Engineering Guidelines for Planning and Designing Hydroelectric Developments," Volume 5, Pumped Storage and Tidal Power, 1989.

EPRI, "Dynamic Operating Benefits of Energy Storage," EPRI AP-4875, October 1986.

EPRI, "Pumped-Storage Planning and Evaluation Guide," EPRI-G5-6669, January 1990.

EPRI, "Technical Assessment Guide, Electricity Supply-1989," EPRI P-6587-L, September 1989.

Koebbe, R.S., "Benefits and Applications of Modular Hydroelectric Pumped Storage," Volume 1, Proceedings, Waterpower '91 Conference, July 1991.

Use Of Weirs For Tailwater Improvements Downstream From TVA Hydropower Dams

Gary E. Hauser[1], William D. Proctor[2], and Tony A. Rizk[3]

Abstract

TVA is designing and constructing innovative weirs below certain of its hydropower dams to: to improve minimum flows between generating periods; and to increase tailwater dissolved oxygen (DO) content during generation. High-performance weirs represent one of several alternatives considered at each dam, as part of TVA's Lake Improvement Plan, which will implement minimum flow and aeration technologies to improve the releases of 16 dams over the next five years. Target minimum flows are sustained by slow drainage of the weir pool through low-level pipes with float-actuated valves that maintain a constant flow when the upstream turbine is not operating. These weirs achieve aeration as overflow nappes impinge on plunge pools during turbine generation in a process similar to natural waterfalls. This poster presentation updates water resource professionals on the status and performance of TVA weir projects.

Introduction

TVA has been involved in design, construction, and performance testing of weirs since 1984. An experimental

[1]Senior Engineer, TVA Engineering Laboratory, P.O. Drawer E, Norris, TN 37828

[2]Engineer, TVA Engineering Laboratory, P.O. Drawer E, Norris, TN 37828

[3]Engineer, TVA Engineering Laboratory, P.O. Drawer E, Norris, TN 37828

gabion reregulation weir constructed in 1984 below Norris Dam on the Clinch River has been meeting minimum flow objectives, but has required several modifications since construction to preserve its structural integrity and performance. An aerating labyrinth weir completed in December 1991 below South Holston Dam on the South Fork Holston River has been meeting or exceeding aeration and minimum flow objectives with minimal operation and maintenance concerns. A timber crib extension to an existing ogee weir, designed to eliminate a recirculating downstream "keeper", was completed in May 1992 on the South Fork Holston River and is performing successfully. An aerating infuser weir completed in November 1992 below Chatuge Dam on the Hiwassee River has also shown promising aeration results in preliminary testing. These weirs and their performance results are summarized in the following sections.

Norris Experimental Reregulation Weir

The Norris tailwater weir is a 1.5-m high rock-filled gabion weir with an impermeable, vertical membrane near the upstream face. The weir was designed to sustain a constant minimum flow of 5.7 m^3s^{-1} between turbine generating periods through a series of low level pipes fitted with float-actuated valves for flow control. In cross section, the weir steps down from headwater to tailwater in a shape that was designed in physical modeling studies to prevent the dangerous "roller" normally associated with low-head weirs. The weir worked well for a period of time and overflow conditions were safe in full turbine flow (240 m^3s^{-1}). After a few years of operation, the gabion wire baskets began to deteriorate. To prevent further erosive degradation of the weir, a concrete cap was installed to preserve the gabions. This change altered flow conditions, causing the dangerous roller to occur downstream. Further physical modeling of the capped gabion structure showed that simple addition of a porous timber crib extension on the downstream face of the weir would eliminate the roller. A 2.7-m (upstream to downstream distance) timber crib extension was constructed and overflow conditions were again safe. Over time, several low level pipes that help provide minimum flow when the weir pool drains became clogged with sediment. Also, trash screens in front of the pipes have required frequent cleaning due to clogging by algal growths that built up on the screens. Currently, the weir operates with the screens removed and with the float-actuated valved pipes locked in fully open

position to make up for the pipes that have become clogged by sediment. TVA is planning to completely replace this weir within the next two years with a timber crib weir using fewer, larger drain pipes and valves to minimize maintenance requirements.

Bristol Weir Timber Crib Extension

The Bristol weir is a 1.3-m high concrete ogee crest designed to provide storage for the municipal water supply. A recirculating flow (submerged hydraulic jump) and deep scour hole existed downstream of this weir during overtopping that was hazardous to river enthusiasts who were using the tailwater in increasing numbers for recreation. The water surface drop across the weir during turbine operation upstream was less than 0.5 m, so the hazard was exacerbated by the innocuous appearance of the flow condition. An extension was added to this concrete water supply weir to eliminate the recirculating flow . Various solutions were tested with a scale physical model, seeking a configuration that avoided recirculation over the range of flows experienced by the prototype, including transition flows during turbine startup and shutdown. The recommended design after this investigation was to fill the scour hole with concrete and build a stepdown timber crib weir extension on the downstream side of the concrete weir. A six-step timber crib 3-m long in upstream to downstream section dimension was constructed. This extension has successfully altered the water surface profile at the Bristol weir from a diving jet with a submerged hydraulic jump into a surface jet with mild standing waves that is completely free of recirculation.

South Holston Aerating Labyrinth Weir

A labyrinth weir has an extended crest length with a repetitive "W" shape in plan view. The extended crest reduces the unit discharge during generation. This reduces the intensity of downstream recirculation to a level that is not hazardous and at the same time increases the aeration. The South Holston labyrinth has a 650 m overflow crest length, an average height of just over 2 m, and a drop height (headwater to tailwater) of 1.4 m during turbine generation. Slotted concrete piers and leveling pad provide the foundation support for this weir, and tongue-and-groove treated timbers placed horizontally in the piers provide the weir walls. Water

overtops the long crest of the weir during generation, creating thin waterfalls that entrain air as the nappes impinge on the plunge pool, creating small bubbles in a turbulent flow field. Labyrinth weirs provide excellent aeration and have higher overflow capacities than conventional weirs, producing less backwater on upstream turbines and less property inundation during high flows. The TVA labyrinth weir, now in operation, is successfully sustaining a minimum flow within about 10 percent of the target 2.5 m^3s^{-1} for 19 hours without recharging the weir pool. The weir is also increasing tailwater DO content during generation (68 m^3s^{-1}). In preliminary testing during January and February 1992, the labyrinth weir aerated over 60 percent of a 2 mg L^{-1} DO deficit in the South Holston Dam releases. It was not possible to test aeration at this weir in the fall 1992 low DO season due to a turbine outage. However, projections based on the preliminary tests indicate that DO at the weir should increase from its incoming value of 3 mg L^{-1} to over 7 mg L^{-1} downstream of the weir during the lowest DO period (saturation DO = 9.5 mg L^{-1}).

Chatuge Aerating Infuser Weir

Research on aerating weirs at TVA since the labyrinth weir has focused on more compact designs that can be constructed with less capital investment. The Chatuge weir, dubbed the "infuser" by TVA, is a hollow, broadcrest weir with specially designed transverse openings in its crest that allow a series of water curtains to fall through to a plunge pool beneath the crest. The Chatuge infuser has a 30 m overflow crest length, an average height of 2.5 m and a drop height (headwater to tailwater) of 1.85 m. This weir is capable of aeration efficiencies similar to that of the labyrinth at about half the construction cost, due to its compact size and shape. During low DO season, the weir is expected to recover about two-thirds of the DO deficit, based on preliminary tests in November 1992. Thus, during the low DO season when DO above the weir is around 1 mg L^{-1}, DO below the weir is expected to be over 6 mg L^{-1}. Partial clogging of the grated infuser deck was observed during the fall leaf drop season, requiring frequent cleaning. Advantages of this weir are low cost and compactness, and disadvantages include higher backwater and property inundation during floods (less efficient crest than labyrinth) and potential clogging of the infuser deck.

Developmental features and performance of all the above weirs are briefly described in the poster presentations. The presentation emphasizes how weirs represent an important alternative for the hydropower utility that can be used to enhance the value of the hydro tailwater for a range of uses while minimizing hydropower impacts, unsafe hydraulic conditions, and environmental disturbance.

Transient Pulses of Chemical Oxygen Demand in Douglas Reservoir

Richard J. Ruane (AM)[1]

Introduction

In August 1992 TVA was experimenting with an oxygen diffuser system in Douglas Reservoir, approximately 0.2 miles upstream from the dam. The project engineer in charge of the evaluation of the oxygen diffuser system called to report that the increase in dissolved oxygen (DO) that he had observed over the past several weeks was decreasing from 2 mg/L to 1 mg/L. A few days later he called again to report that the DO increase had dropped to only 0.4 mg/L. This paper summarizes the results of an investigation to determine the cause for the drop in DO increase. It was determined that the DO was consumed by chemical substances that occurred in the water that was near the bottom of the reservoir. This mass of water moved through the forebay region over a period of several weeks from upstream sources. Samples of the water were analyzed for chemical constituents and also evaluated for the settling characteristics once the water was oxidized. The design for the full-scale oxygen diffuser system was adjusted to accommodate this short-term oxygen demand, but studies are continuing to determine the sources of it and to consider oxidization of this material at upstream locations. Although it is anticipated that the oxygen diffuser system can satisfy the oxygen demand, the precipitates that occur upon oxidation of this material may result in excessive accumulation of precipitates in the forebay and also in generator cooling water systems.

Background

Douglas Reservoir is located on the French Broad River at river mile 32.4 and has a drainage area of 4541 square miles, with only a small portion of the drainage area controlled by upstream reservoir projects. At normal summer pool, the reservoir has a length of 43 miles, a volume of 1,408,000 acre-feet, and a surface area of 30,400 acres. The average annual reservoir flow is 6750 cfs, resulting in a nominal hydraulic retention time of 104 days. The generating capacity at Douglas is 121 megawatts, accomplished with 4 turbine units. The depth at the dam is about 125 feet at normal summer pool level and the depth of the invert of the penstock is 112 feet.

The minimum DO discharged from Douglas is typically close to 0 mg/L. The DO in the turbine discharges from Douglas drops rapidly during the spring and decreases below 4 mg/L during the month of June and recovers to 4 mg/L during the month of

---

[1]Senior Environmental Engineer, Water Management, Tennessee Valley Authority, Chattanooga, TN 37402

October. The lowest DO period is from about mid-July to the beginning of September when the DO is less than 1 mg/L. From a limnological viewpoint Douglas is considered to be a transitional reservoir (Ruane and Hauser, 1990 and 1991) having some characteristics similar to mainstem projects but also having some characteristics similar to storage projects. As a result Douglas experiences relatively strong stratification in the first part of the summer but very weak stratification toward the end of the summer and into the fall. Because of the relatively shorter retention time compared to storage reservoirs, the temperatures in the bottom waters of Douglas increase much more rapidly than those in storage impoundments, and the DO is consumed by decomposition of organic matter much more rapidly, thus resulting in the low DO beginning earlier in the summer. Low DO persists even during periods of weak thermal stratification because it is sufficient to prevent mixing from the top of the reservoir to the bottom, and, therefore, DO can be significantly stratified.

The inflows to Douglas are known to contain high concentrations of organic matter and nutrients from nonpoint sources and upstream industrial wastewater discharges. As a result, the DO in the water in Douglas is not only consumed, but anaerobic bacteria start decomposition of the organic matter in the absence of oxygen and begin to consume the oxygen that is included in compounds such as iron-, manganese-, sulfur-, and nitrogen-oxides. When the oxygen is taken from these constituents, the byproducts are called anoxic products and include such constituents as ferrous iron, dissolved manganese, hydrogen sulfide, ammonia, and dissolved phosphorus. If oxygen is introduced in the presence of these chemical constituents, they will oxidize and consume available DO, some of them (i.e., ferrous iron and hydrogen sulfide) oxidizing rather rapidly within minutes and hours (Johnson et al., 1991).

TVA plans to increase the DO in the releases from Douglas to 4 mg/L. About half of the DO increase will be provided by surface water pumps that will pump water from the epilimnion down to the vicinity of the turbine intakes such that it will be drawn through the turbines. About one-third of the turbine flow will then be provided from water in the epilimnion. The other method to be used to achieve the 4 mg/L will be the oxygen diffusers that were being tested during August 1992.

During 1992 another program within TVA was investigating the various problems that had been expressed about Douglas Reservoir, such as excessive eutrophication; low fishery counts; high concentrations of organic materials in the upper end of Douglas Reservoir; high turbidity in the bottom waters of the reservoir and the discharge from the dam; and dissolved iron and manganese, ammonia, and hydrogen sulfide in the reservoir and its tailwater. These studies involved monthly monitoring throughout the reservoir but also bi-weekly monitoring in the forebay. Even with this amount of monitoring the transient pulse of the high chemical oxygen demand would not have been adequately addressed without a special study conducted at the time when the chemical oxygen demand was at its greatest. Hence a special study was conducted for a 2-day period on September 2 and 3, 1992, and a smaller study on September 11. These studies focused primarily on the chemical oxygen demand but also involved some analyses of parameters that may help explain the cause of the chemical oxygen demand as well as the behavior of the water after it was oxidized.

Field Study

To determine the potential for chemical oxygen demand, a profile of the water quality in the water column was obtained about 0.1 miles upstream from the diffuser system. Field parameters were obtained using a Hydrolab Surveyor II to measure DO,

temperature, conductivity, pH, and oxidation reduction potential. Turbidity was determined in the field using a Hach Turbidimeter (model number 2100P). Hydrogen sulfide was determined using a field procedure described elsewhere (Johnson, Ruane, and Howe, 1991). Samples were collected for laboratory analyses at various depths for the following parameters: total and volatile nonfilterable residue, organic nitrogen, ammonia, nitrite-plus-nitrate, total phosphorus, total organic carbon, calcium, manganese, chloride, sulfate, copper, total and dissolved iron, total and dissolved manganese, zinc, total aluminum, total silicon, and true and apparent color. In addition, a portion of the water was aerated and the DO demand was measured over a short period of time (about 2 hours) until the demand stabilized. The results of the DO demand studies are presented in figure 1 and represent the characteristics of the water that was 1-, 2-, and 4-meters above the bottom of the reservoir for the dates September 2, 3, and 11.

It was noted during the field studies that the water collected from 1 and 2 meters above the bottom of the reservoir was black in appearance, on both the filter pads as well as in a white bucket of water collected from the near-bottom layers. The samples collected 4 meters above the bottom had a light brown (tan) color. Considering the apparent high concentrations of oxygen-demanding material and the color of the water from the near-bottom of the reservoir, it was decided on the first sampling date to collect approximately 30 gallons of water to return to the TVA Environmental Engineering Unit Operations Lab in Chattanooga to conduct settling studies. The 30-gallon sample was collected in a manner so as to minimize aeration but apparently some oxygen was available to the sample as indicated by the change in color overnight to reddish-brown.

Discussion

The short-term DO demand (DOD) was observed mainly in the near-bottom waters at sampling depths of 32 and 33 meters. On September 2 the average DOD at these two sampling points was about 1.7 mg/L and had dropped appreciably to about 1 mg/L by the next day. The short-term DOD on September 11 was slightly less than the DOD measured on September 3. The water collected from these lower two depths was black in appearance on all three dates. The water collected from the 30 meter depth had a short-term DOD only on September 2 and was about 0.5 mg/L. Considering that the oxygen diffuser system was located at an elevation of about the same depth as the samples collected from the 32- and 33-meter depths, it is readily apparent that the drop in DO increase that had been observed for the oxygen diffuser system can be explained by the short-term DOD. The travel time of water in the reservoir between the pilot-scale oxygen diffuser system and the turbine outlets was greater than 2 hours during the diffuser evaluations discussed above.

To help determine the cause for the short-term DOD as well as the source of the material causing the short-term DOD, various water quality constituents were determined in the TVA Environmental Chemistry Laboratory. The results of these analyses on selected constituents that were found to be pertinent to the short-term DOD are summarized in table 1. The most apparent observation that can be made (figure 2) from this table is that the near-bottom samples had the highest concentrations of iron, manganese, sulfide, ammonia, total phosphorus, silicon, total organic carbon, and volatile nonfilterable residue (suspended solids), as well as the lowest concentrations of sulfate and nitrate-plus-nitrite. These patterns are consistent with limnological findings in natural lakes that are considered to be hypereutrophic (Wetzel, 1983), that is lakes experiencing excessive loads of organic matter. The data shown in table 1 indicate a significant presence of excessive organic matter as well as activity of anaerobic bacteria consuming oxygen from metal oxides, sulfate, and nitrate-plus-nitrite. The end product is a collection of water quality constituents

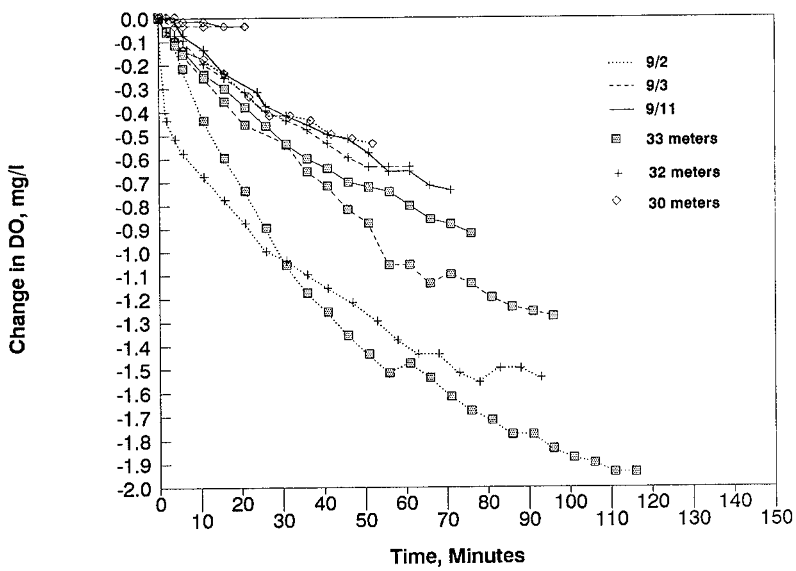


Fig. 1 Short-term oxygen demand in Douglas forebay

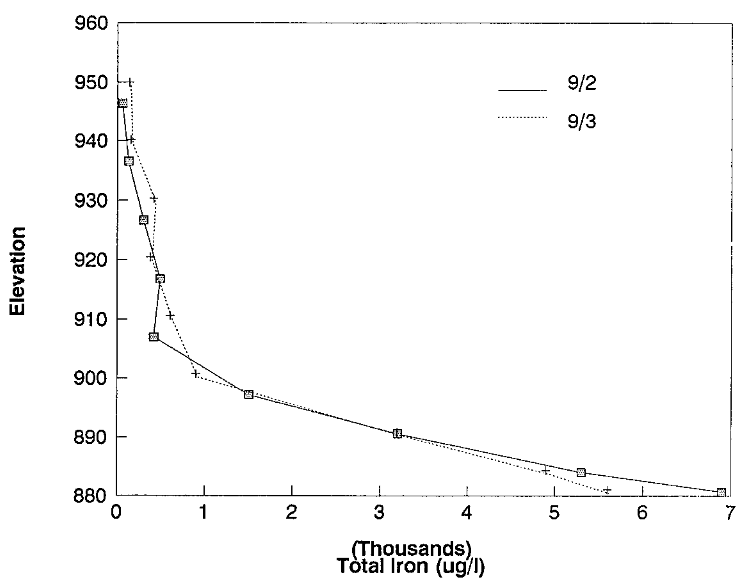


Fig. 2 Iron concentrations in Douglas forebay (surface elev., 989)

Table 1. Characteristics of Water Quality in the Lower Depths of Douglas Forebay During the Transient Pulse that Occurred in late August and early September 1992

DATE	DEPTH (M)	WATER TEMP CENT	CNDUCTVY FIELD MICROMHO	PH SU	IRON FE, DISS UG/L	IRON FE, TOT UG/L	MANGNESE MN, DISS UG/L	MANGNESE MN UG/L	SULFIDE TOTAL UG/L	SULFATE SO4-TOT MG/L
September 2	33	17.2	171	7.10	-	6900	-	1100.0	58.50	2
	32	18.1	160	7.10	5300	5300	930.0	930.0	26.00	4
	30	19.3	150	6.70	3100	3200	800.0	800.0	14.30	6
	28	20.2	141	6.40	1300	1500	670.0	670.0	<1	7
	25	21.7	134	6.40	20	420	290.0	290 0	<1	10
	22	22.5	155	6.40	70	490	240.0	240.0	<1	13
	19	23.1	155	6.40	40	300	180.0	180.0	<1	12
	16	23.9	163	6.50	10K	130	120.0	120.0	<1	11
	13	24.7	149	6.70	10K	60	60.0	84.0	<1	12

DEPTH (M)	AMMONIA TOTAL MG/L	NO2&NO3 N-TOTAL MG/L	PHOS-TOT MG/L P	SILICON SI, TOT UG/L	ALUMINUM AL, TOT UG/L	T ORG C C MG/L	RESIDUE VOL NFLT MG/L	RESIDUE TOT NFLT MG/L	TURBDITY FIELD NTU
33	1.600	.01K	.600	6000	170	4.7	5	6	6.2
32	1.300	.01K	.480	5500	160	4.0	4	4	5.5
30	.860	.01K	.250	5000	210	3.3	6	7	7.3
28	.590	.01K	.100	4700	180	2.7	3	8	7.6
25	.220	.15	.020	4100	210	2.8	3	5	8.7
22	.110	.52	.040	4200	300	3.3	3	6	9.8
19	.080	.48	.030	4000	210	2.8	3	4	5.5
16	.040	.42	.030	3500	100	2.8	3	3	3.3
13	.060	.16	.030	2600	70	2.4	2	3	3.3

DATE	DEPTH (M)	WATER TEMP CENT	CNDUCTVY FIELD MICROMHO	PH SU	IRON FE,DISS UG/L	IRON FE,TOT UG/L	MANGNESE MN, DISS UG/L	MANGNESE MN UG/L	SULFIDE TOTAL UG/L	AMMONIA N TOTAL MG/L	NO2&NO3 N-TOTAL MG/L	SILICON SI, TOT UG/L	ALUMINUM AL, TOT UG/L	TURBDITY FIELD NTU
September 3	33	17.5	175	7.00	5300	5600	940.0	960.0	47.50	1.300	.01K	5800	250	7.6
	32	18.6	162	6.80	4600	4900	780.0	920.0	18.00	1.200	.01K	5500	220	5.8
	30	19.5	153	6.70	2800	3200	780.0	840.0	6.10	.890	.01K	5100	340	6.4
	27	20.9	147	6.40	190	900	610.0	610.0	<1	.520	.01K	4600	230	13.2
	24	22.3	157	6.50	60	610	250.0	300.0	<1	.240	.32	4400	340	13.1
	21	22.6	162	6.50	70	380	96.0	120.0	<1	.020	.66	4100	200	6.2
	18	23.1	159	6.50	40	420	51.0	140.0	<1	.090	.40	3900	190	6.2
	15	24.0	166	6.50	10	160	51.0	51.0	<1	.040	.45	3400	120	3.4

called anoxic products that exert a DOD. Iron and sulfide will oxidize on the-order-of-minutes and hours, whereas manganese, ammonia, and phosphate exert a DOD over a period of days and weeks. However, it should be noted that iron and sulfide form a precipitate that is not measured using the procedure for analyzing sulfides so, therefore, the results do not reflect the particulate that probably was present in the form of iron sulfide. Iron sulfides are commonly known by limnologists to have a black appearance and are also known to cause a black "film" to cover sediments. The decrease in nitrate-plus-nitrite in the near-bottom water is not unusual in many reservoirs experiencing low DO; however, the reduction in sulfate in the near-bottom water is especially indicative of hypereutrophic conditions.

Considering the data in table 1 and the short-term DOD results, it is obvious that iron sulfides were present at the lower 2 sampling points, i.e., at the 32- and 33-meter sampling depths. This conclusion is based on the lower values of sulfate measured at the 32- and 33-meter depth locations as well as the black color of the water samples as mentioned earlier. It is also apparent from the data that the short-term DOD on September 2 at the 30 meter depth was attributed entirely to dissolved iron, because the observed dissolved iron concentration would have required stoichiometrically the observed short-term DOD to satisfy the oxidation of ferrous iron to ferric iron. At the lower 2 sampling depths the measured dissolved iron concentrations were not sufficiently high enough to cause the observed short-term DOD, so it is concluded that the remainder of the short-term DOD (about half of the observed DOD) was attributed to the oxidation of iron sulfides.

A comparison of the data reported in table 1 with data collected from the monitoring program for the other purposes as discussed earlier indicates that the water mass observed on September 2 and 3 was a transient pulse. Compared to water quality data collected on August 24 in the forebay (only 9 to 10 days before the September 2 and 3 event) the data on the near-bottom samples on September 2 and 3 were greater as indicated by the following multiplication factors: dissolved iron, 3x greater; dissolved manganese, 1.5x; hydrogen sulfide, 20x; ammonia, 2x; total phosphorus, 2x; and TOC 1.5x. The transient nature of bottom water in Douglas Reservoir becomes apparent by examining figure 3. This figure summarizes the temperature and dissolved iron observations for sampling dates on August 11 and September 9, 1992. The figure summarizing the temperature data on these two dates indicates that 2 types of density currents occurred over this 29-day period. One type of density current occurred as result of the hydropower generation at Douglas as indicated by the drop in elevation of the temperature isopleths in the forebay, i.e., the 22-degree isopleth decreased in elevation from 940 to 900 as a result of the volume of cool bottom water being released through the turbines. By tracing the 22 degree isopleth on September 9 to upstream regions of the reservoir, it is apparent that the drop in elevation for this isopleth at upreservoir locations lagged in time behind the drop in elevation at the forebay. This condition causes density currents, i.e., the cooler water that is 22 degrees at river mile 55 is in a region of the reservoir that would otherwise be about 24 degrees and the 22-degree water which is denser will flow along the original river channel towards the dam. As indicated by the figure, the 22 degree isopleth on September 9 is inclined along the channel bottom of the reservoir, and this is indicative of density currents. The other type of density current that was occurring on September 9 and probably had for at least several days prior to September 9 is indicated by the 24 degree isopleth at the upper end of the reservoir. This isopleth is indicative of inflow water plunging below the surface waters and flowing along the bottom of the reservoir as is indicated in the figure. The 24-degree isopleth is inclined along the reservoir bottom channel on September 9 whereas on August 11 the 24-degree isopleth was essentially level throughout the reservoir at elevation 950. The lower graph in figure 3 illustrates the

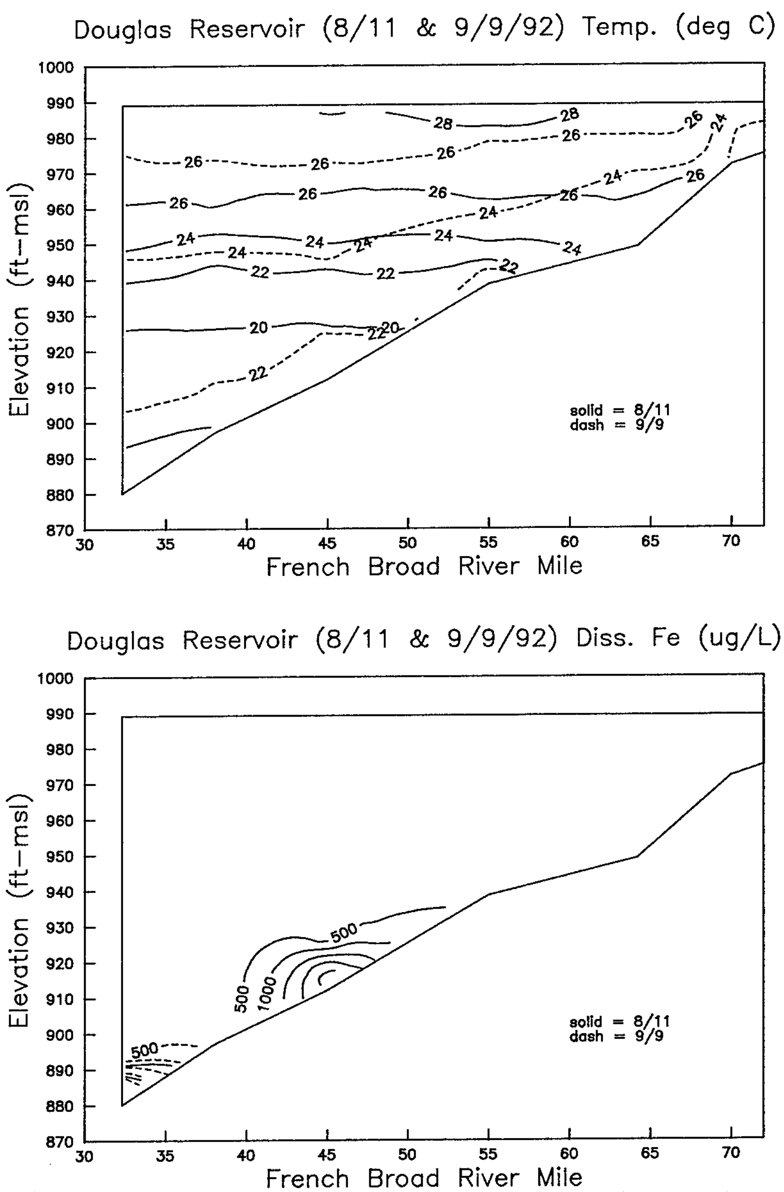


Fig. 3 Longitudinal depictions of temperature and dissolved iron in Douglas Reservoir on Aug. 11 and Sept. 9, 1992

movement of dissolved iron between these two sampling dates and vividly illustrates the dramatic movement of dissolved iron between August 11 and September 9. (It should be noted that while the peak concentrations of dissolved iron on August 11 were seen at river mile 45, the original source of the dissolved iron was further upstream a few weeks earlier in the summer--somewhere around river mile 55.)

The other indications of the transient nature of the anoxic products can be obtained by comparing data from the forebay on September 2 and 3 with data collected 13- and 23-miles upstream on August 11 and July 27, respectively. The increases in the forebay were as indicated by the following multiplication factors: dissolved iron, 2x; dissolved manganese, about the same concentration; hydrogen sulfide, 30x; ammonia, 1.7x; total phosphorus, 2x; and total organic carbon, 1.4x. These increases between the upstream stations and the forebay were caused to some extent by the production of anoxic products by anaerobic bacteria over the 3- to 5-week period. But some of the constituents appeared to experience an excessive increase in the forebay if the increases between river miles 55 and 45 are used to predict the constituent concentrations that would be expected in the forebay. These excessive increases included iron, ammonia, total phosphorus, and total organic carbon. In fact, the forebay concentrations of these constituents measured on September 9 were more in line with expectations, indicating again the transient nature of the water mass observed in the near-bottom water on September 2 and 3.

It is hypothesized that scour of a thin layer of chemical precipitates (i.e., iron sulfides) and organic matter explains the excessive concentrations observed on September 2 and 3. This hypothesis is based on the following supporting evidence: 1) the volatile solids content of the total suspended solids was about 80 percent whereas the volatile residue associated with sediment samples of the top 2 or 3 centimeters of sediment at various locations in the reservoir are only 10 percent; 2) a comparison of pore water concentrations from upstream sediments having the same water quality constituents as in the near-bottom water on September 2 and 3 indicates that increases in total organic carbon and iron were relatively higher than the increases in ammonia and manganese; and 3) it would be expected that the surface of the sediments would contain higher concentrations of iron, sulfides, and organic material.

The 30-gallon sample that was collected to evaluate settling characteristics was placed in a 10-inch diameter settling column, and the water was aerated at the beginning of the study to attempt to simulate the effects of the oxygen diffuser system in increasing DO. Two settling regimes were evident. In the first 24 hours, large particulates formed (having a reddish black and reddish brown appearance) and settled to the bottom of the settling column. During this initial period of rapid settling about 50 percent of the total suspended solids, manganese, aluminum, and total phosphorus were removed from the water column, but only 5 percent (about .5 mg/L) of the total iron precipitated. As settling continued over the next 2 weeks, total iron was reduced by 36 percent and the other constituents were essentially 100 percent removed from the water column. Volatile organic matter was reduced about 80 percent in the 2 weeks, but about half of it may have been decomposed. This phenomenon is common for near-bottom waters of hypereutrophic lakes as explained by Wetzel (1983) and the following discussion is taken from him. When ferrous iron is oxidized, a very finely divided precipitate of ferric hydroxide may form that has colloidal properties. Colloidal particles of ferric hydroxide are commonly positively charged. Other ions in solution, negatively charged clay particles, organic colloids and other suspended solids can neutralize the charges of the hydroxide colloidal particles. The uncharged aggregates join to form a rapidly settling precipitate. Other metals can be adsorbed by and coprecipitated with the ferric hydroxide precipitate.

Conclusions

The design of aeration systems for some hydropower reservoirs will need to consider short-term DOD which can occur in transient pulses. These conditions are more likely to occur in reservoirs that are considered to be transitional type projects, i.e., those projects with nominal retention times of between 25 and 200 days of retention time, have low-level outlets, and receive excessive amounts of nutrients and organic matter in the inflows to the reservoir. TVA has designed and is installing an oxygen diffuser system at Douglas in a manner so as to accommodate the excessive short-term DOD. However, after the system is installed, monitoring will be conducted to determine the effectiveness in meeting the DO objectives as well as to determine the best operational scheme in conjunction with the surface water pumps. The studies will also determine the effects of the precipitates and the possible excessive accumulation of these precipitates in the forebay and in the cooling water systems for the generators.

Work is also planned to determine the location within the reservoir from which these transient pulses originate and to determine whether small aeration systems may be appropriate to minimize the formation of anoxic products or at least cause the precipitates to accumulate further upstream in the reservoir.

Additional assessments are planned to determine the nature and sources of the excessive organic matter in Douglas in hope of using the information to encourage the reduction of contamination in the watershed from nonpoint sources and possibly industrial sources.

Appendix

References

Johnson, J.T., R.J. Ruane, and L.H. Howe III, 1991, "Hydrogen Sufide in Hydropower Reservoirs," Proceedings of the ASCE Waterpower '91 Conference, Denver, CO.

Ruane, R.J., and G.E. Hauser, 1990 "Factors Affecting Water Quality in the Releases from Hydropower Reservoirs," Proceedings of the 52nd American Power Conference, Illinois Institute Technology, Chicago.

Ruane, R.J., and G.E. Hauser, 1991 "Factors Affecting Dissolved Oxygen in Hydropower Reservoirs," Proceedings of the ASCE Waterpower '91 Conference, Denver, CO.

Wetzel, R.G., 1983, Limnology, 2nd ed., Saunders College Publishing, Philadelphia.

Acknowledgements

The author was assisted in field studies by Phil Clapp, Sam Grindstaff, and Bruce Coakley. The settling studies were performed under the direction of Tina Tomaszewski. Chemical analyses were under the direction of Bill Scott. Ellen Hammond and Kathy Winters were responsible for data management and prepared graphs. Kathy Standifer and Janice Dockery prepared the manuscript.

Determining the Number of Transects in IFIM Studies

Alan W. Wells,[1] M. Elizabeth Conners,[2] Susan G. Metzger,[3]
and John Homa, Jr.[4]

Abstract

In designing Instream Flow Incremental Methodology (IFIM) studies, two important considerations are the representativeness of the results and the cost of the study. Both are largely determined by the number of transects sampled. To design a cost-effective sampling program for a new study, data from a previously sampled river were used to obtain bootstrap estimates of the 95% confidence intervals that would be expected to arise from sampling various numbers of transects. The procedure suggested that 10 to 15 transects per river segment was a reasonable compromise between cost and statistical precision for species occupying more than 50% of the wetted area. For less widespread organisms more than 20 transects would be required. These results and methods may serve as useful guidelines for designing future IFIM studies.

Introduction

Increasingly, the licensing or relicensing of small hydroelectric facilities requires the application of Instream Flow Incremental Methodology

[1]Program Manager, Lawler, Matusky & Skelly Engineers, One Blue Hill Plaza, P.O. Box 1509, Pearl River NY 10965

[2]Senior Ecologist, Ichthyological Associates, Inc., 50 Ludlowville Road, Lansing, NY 14882

[3]Director, Environmental Sciences Section, Lawler, Matusky & Skelly Engineers, One Blue Hill Plaza, P.O. Box 1509, Pearl River, NY 10965

[4]President, Ichthyological Associates, Inc., 50 Ludlowville Road, Lansing, NY 14882

(IFIM) (Bovee 1982, 1986) procedures to predict changes in riverine habitat that may result from hydroelectric operations. As part of the IFIM process, hydrodynamic characteristics of streams are related to habitat suitability through a collection of Physical Habitat Simulation programs (PHABSIM). The primary output from these programs is an estimate of the amount of usable physical habitat, called the *weighted usable area* (WUA), under various flow conditions. A plot of WUA vs streamflow can then be used to select the flow condition that maximizes habitat for a particular objective.

Obtaining the data necessary for the calculation of the WUA usually requires a detailed field survey. Although the actual procedure may vary somewhat from study to study, depending on objectives and models used, there are several common elements. After the stream region to be surveyed has been selected, transects are established perpendicular to the streamflow. Physical habitat characteristics are then measured at intervals along each transect. Typically, these habitat characteristics are current velocity, water depth, and substrate/cover, but other measures may be included.

Sampling transects is a labor-intensive effort, and total sampling costs can easily become prohibitive as the number of transects increases. Therefore, it is desirable to sample as few transects as possible while obtaining sufficient data to characterize the water body with reasonable accuracy and precision.

The placement of transects, and to some extent the number of transects, is related to the hydraulic model used in the analysis. Early IFIM studies often used the Water Surface Profile (WSP) hydraulic simulation model. This model requires transect placement in each of the hydraulic controls, i.e., a change in the slope of the water surface, in addition to those in each distinctive habitat type. This approach can require a large number of transects.

Morhardt et al. (1983) suggested using an alternative hydraulic model, the IFG4, in conjunction with habitat mapping to increase the accuracy of IFIM predictions while reducing the overall cost. This procedure does not require the placement of transects at hydraulic control points. Instead, the water body is divided into strata of homogeneous habitat types, e.g., pools, riffles, and runs, and mapped. Transects need only to be placed within these strata. Morhardt et al.'s method clearly reduces the overall number of transects necessary for many IFIM studies but leaves unaddressed the question of how many transects are necessary within each stratum to adequately characterize the water body. This paper proposes a methodology for addressing the sample size issue and offers some guidelines for an appropriate number of transects.

Methods

Physical habitat within a water body can be quite variable from transect to transect due to differences in such factors as channel shape, substrate, cover, and bank characteristics. If only a few transects are studied, these *n* transects may or may not be representative of the water body as a whole. Repeating the same study with another selection of *n* transects may produce markedly different results. As the number of transects increases, however, the coverage will become more representative. Repeated studies using a very large number of transects should yield nearly identical results. The degree to which study results can be repeated, i.e., the sampling precision, is expressed by the standard deviation of the mean WUA.

Given the desired degree of sampling precision and the standard deviation of the mean (standard error), the requisite number of samples can be calculated easily using standard methods such as those found in Cochran (1977). However, in designing an IFIM study several difficulties render these methods inappropriate. First, the target stream likely has not been previously subjected to IFIM studies. There are thus no data from which the standard deviation of the mean WUA can be calculated. Second, the complex functions used to translate the physical habitat to habitat suitability, i.e., Habitat Suitability Indices (HSI), may yield decidedly non-normal (Gaussian) distributions of WUA estimates. The standard methods typically require normally distributed errors, especially for small sample sizes.

In an effort to design a cost-effective IFIM sampling program for a previously unsampled stream in the central United States, we devised the following procedure based on a bootstrap sampling experiment (Efron 1979; Efron and Tibshirani 1986). Empirical data, taken from 21 transects, were available from an eastern United States stream of moderate gradient. To ensure that this surrogate stream would have sufficient habitat variability to represent the possible habitats found in the target stream, we included six transects from pools, 10 transects from riffles, and five transects from runs. For each of the 21 transects the WUA for longnose dace adult habitat, fallfish spawning habitat, white sucker fry habitat, mayfly (*Stenonema* spp.) habitat, and total wetted area was calculated and stored in a data file.

For each species 500 random samples, with replacement, of *n* number of transects were drawn from the stored WUA data. The procedure was repeated for all *n* transects, one to 21. The standard deviation (SD) of the 500 bootstrap samples yields the expected sampling variability. Multiplying the SD by 1.96 gives the approximate 95% confidence interval (CI). For comparison among species and life stages the CI was divided by

the bootstrap sample mean. This value is referred to as the relative width of the 95% CI.

Results

The results of the analysis for total wetted area, longnose dace adult habitat, fallfish spawning habitat, white sucker fry habitat, and mayfly habitat for streamflows of 120 cfs are presented in Figure 1. Results were also computed for flows of 20, 40, 60, 80, 100, 140, 160, 320, and 625 cfs. The results at these flows were similar to those at 120 cfs.

The relative width of the 95% CI of WUA decreased in an approximately exponential fashion as the number of transects increased. For longnose dace and mayfly a sample size of 15 transects yielded a relative width of less than 20%. Of the four species, mayflies displayed the best precision: one, five, 10, and 15 transects yielded a precision of ±62, 27, 19, and 15%, respectively. Overall, total wetted area displayed the best precision; 15 transects yielded a precision of approximately ±10%, 21 transects a precision of approximately ±9%.

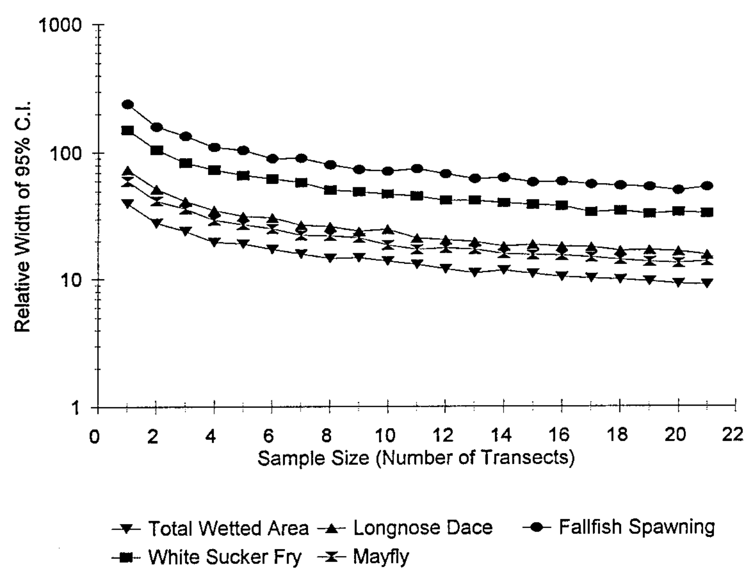


FIGURE 1. - Relative Width of 95% Confidence Interval With Number of Transects at Flow of 120 cfs.

It should be noted that the precision of the estimate is highly influenced by the percentage of habitat covered (Figure 2). The total wetted area represents the entire wetted stream area, only part of which may be suitable habitat for fish and invertebrates. Species and life stages with very specific habitat requirements may occupy only a small proportion of the total wetted area, while species and life stages with less stringent requirements may occupy a large proportion. The larger the proportion of wetted habitat used by a target organism, the greater the likelihood of being sampled and the greater precision for any given sample size. For example, longnose dace and mayfly use more than 60% of the total wetted area. The precision resulting from 21 transects for these two species was ±16 and ±14%, respectively. In contrast, habitat for fallfish spawning and for white sucker fry is much less common (<20%) and displays an increase in variability. These less common habitats were estimated within ±53 and ±33%, respectively.

Our findings suggested that a sample size of 10 to 15 transects per stratum offers a reasonable compromise between cost and sampling precision for organisms expected to occur in most of the available habitat, i.e., >50% of the total wetted area. However, for organisms with very specific habitat requirements or limited suitable habitat within a given stream, more than 15 to 20 transects may be required.

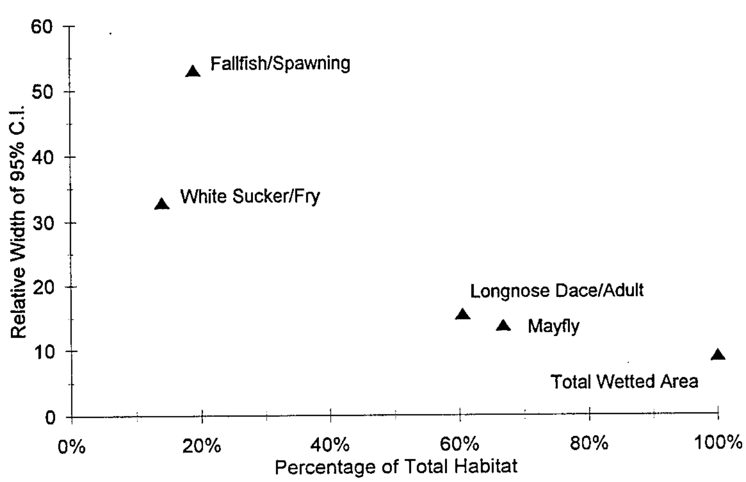


FIGURE 2. - Relationship Between Percentage of Total Wetted Area Occupied and Relative Width of 95% Confidence Interval at Flow of 120 cfs.

References

Bovee, K.D. 1982. A guide to stream habitat analysis using the Instream Flow Incremental Methodology. Instream Flow Information Paper 12. U.S. Fish and Wildl. Serv. Biol. Serv., FWS/OBS-82/26. 248 pp.

Bovee, K.D. 1986. Development and evaluation of habitat suitability criteria for use in the Instream Flow Incremental Methodology. Instream Flow Information Paper 21. U.S. Fish and Wildl. Serv. Biol. Rep. 86(7). 235 pp.

Cochran, W.G. 1977. *Sampling Techniques*. 3rd. ed. New York: John Wiley & Sons. 428 pp.

Efron, B. 1979. Bootstrap methods: Another Look at the Jackknife. The Annals of Statistics, 7:1-26.

Efron, B., and R. Tibshirani. 1986. Bootstrap methods for standard errors, confidence intervals, and other measures of statistical accuracy. Statistical Science, 1:54-74.

Morhardt, J.E., D.F. Hanson, and P.J. Coulston. 1983. Instream flow analysis: Increased accuracy using habitat mapping. Waterpower "83" An International Conference on Hydropower. Conf. Proc. Vol 3: Environmental Impacts, pp. 1294-1304. Tennessee Valley Authority.

SPATIAL SIMULATION OF SMALLMOUTH BASS IN STREAMS[1]

H.I. Jager[2], D.D. Schmoyer[3], M.J. Sale[2],

M.J. Sabo[4], W. Van Winkle[2], D.L. DeAngelis[2]

ABSTRACT

The hydropower industry and its regulators are hampered by the inability to predict the relationship between alternative flow regimes and fish population response. We have developed a spatially explicit, individual-based model of populations of smallmouth bass in streams as part of the Compensatory Mechanisms in Fish Populations Program (see Sale and Otto 1991). In the model, the profitability of alternative stream locations varies in response to habitat depth and velocity through changes in the frequency of prey encounters and the metabolic costs experienced by fish.

We conducted an evaluation of our hydraulic simulation at the scale of individual stream cells. The potential error in predictions for individual cell velocities suggests that larger-scale model predictions for the representative reach are most appropriate. At this scale, the model appears to produce

[1]Research sponsored by the Electric Power Research Institute under Contract No. RP2932-2 (DOE No. ERD-87-672) with the U.S. Department of Energy under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.. The EPRI Project manager is Jack Mattice, P.O. Box 10412, Palo Alto, California, 94303 (415/855-2763). This paper is Publication No. 4043 of the Environmental Sciences Division, Oak Ridge National Laboratory.

[2]Environmental Sciences Division, Oak Ridge National Laboratory, P.O. Box 2008, Oak Ridge, TN 37831-6036 (615/574-8143 or 615/574-7305).

[3]Computing and Telecommunications Division, Oak Ridge National Laboratory, P.O. Box 2008, Oak Ridge, TN 37831-6036 (615/574-5707).

[4]Department of Fisheries and Wildlife Sciences, Virginia Polytechnic Institute and State University, Blacksburg, VA 24061-0321 (703/231-5320).

realistic patterns in the growth and dispersal of young-of-year smallmouth bass. This verification step allows us to proceed with greater confidence in evaluating the original question of how smallmouth bass populations respond to alternative flow regimes.

INTRODUCTION

The use of habitat-based models to predict fish numbers is adequate only when biological feedbacks are unimportant. The U.S. Fish and Wildlife Service's continually evolving Physical Habitat Simulation (PHABSIM) methodology is the principal quantitative tool currently used in the regulation of instream flows. PHABSIM relies on the assumption that the number of fish of a particular lifestage is set by the total amount of "suitable" habitat, which depends only upon streamflow. PHABSIM predictions of the amount of suitable habitat respond instantaneously to changes in flow. The method assumes that fish densities track changes in habitat as they might a time-variable carrying capacity. It is not clear how densities in the field relate to these habitat-based predictions -- is there a point in time when the habitat-based predictions should correspond to actual fish densities? Although suitable habitat is predicted for each different lifestage, the habitat-based methods do not consider biological feedbacks between and within lifestages.

Our model predictions of the number and growth of young-of-year smallmouth bass derive from simulation of the daily activities of individual fish responding to changes in a heterogeneous stream habitat. We borrow the hydraulic modeling component of PHABSIM and superimpose on it a mechanistic model of smallmouth bass reproduction and young-of-year dynamics. The biological model is individual-based and the physical model is spatially explicit (Jager et al., submitted). Abiotic influences on reproduction and growth are modeled through the spatial variation in habitat quality in response to temperature and flow. Biological feedbacks in the model include: (1) the effect of the number of reproductive adults on young-of-year densities, (2) the effect of competition among young-of-year fish for space and food on growth and survival, (3) the effect of prey densities on growth and survival, (4) and the effect of size-dependent predation on the size distribution and number of young-of-year fish that recruit to the first year class.

SIMULATION METHODS

The model is currently being applied to the North Anna River in Virginia. A workstation analysis and visualization environment (PV/WAVE) is used to display the results of hydraulic simulation, the time series of weighted usable area, and the predictions generated by our fish model. These include temporal and spatial patterns in the density, movement, and habitat preferences of simulated fish.

Simulation of Stream Habitat. The daily foraging, growth, survival, and movement of individual smallmouth bass are simulated in a heterogeneous stream habitat that is defined by water temperature, depth, and velocity. Like the habitat-based models, we rely on a representative reach to describe the status of young-of-year smallmouth bass in the stream system of interest. We partition a representative stream reach into spatial cells such that each cell contains one or more measurement stations used in a PHABSIM survey of the reach. These cells provide a fixed spatial reference system for the duration of the simulation. The habitat simulation predicts the spatial distribution of depth and velocity as a function of average daily flow. Local (within-cell) velocity predictions depend on several cell-specific hydraulic parameters that are estimated by PHABSIM, including two parameters of a power function developed between velocity and flow and estimates of Manning's n at each calibration flow. The details of how we predict depth and velocity from flow are given in Jager et al. (submitted).

Simulation of Fish Growth and Survival. The representative reach can be thought of as a collection of cells that vary in growth potential for fish of different sizes. Growth of each individual smallmouth bass is modeled using a standard bioenergetics model (DeAngelis et al. 1991) that subtracts the energetic costs incurred by each fish from its assimilated foraging intake on a daily basis. Both components, prey intake and energetic costs, depend on flow (Jager et al., submitted). Two examples are: (1) prey densities in the model are limited by a habitat-dependent carrying capacity that varies in time and space as the habitat suitability indices (HSI) for prey taxa respond to flow; and (2) the energetic cost of spending time in a particular cell increases with local velocity. These and other relationships that we simulate between the growth and survival of individual fish and habitat are described in greater detail in Jager et al. (submitted).

Simulation of Fish Movement. We assume that fish inhabit one cell per day. Departure from the current day's cell is initiated either by degradation of the local habitat caused by extreme flows (HSI = 0.0) or by insufficient daily growth of the fish. The movement and habitat utilization of model fish can be observed with the help of our PV/WAVE visualization program.

RESULTS AND DISCUSSION

Simulation of Stream Habitat. Predicted velocities are shown for stream cells in a representative reach of the North Anna River for the low calibration flow (1.4 m^3/s) in Figure 1a and for the high calibration flow (8.8 m^3/s) in Figure 1b. The maps in Figure 1 show an overhead view of the stream. Each cell is characterized by velocities predicted at 5 measurement stations that lie within the cell. The measured velocities are shown for comparison in Figures 1c and 1d. Figure 2a compares predicted and observed velocities for these cells over

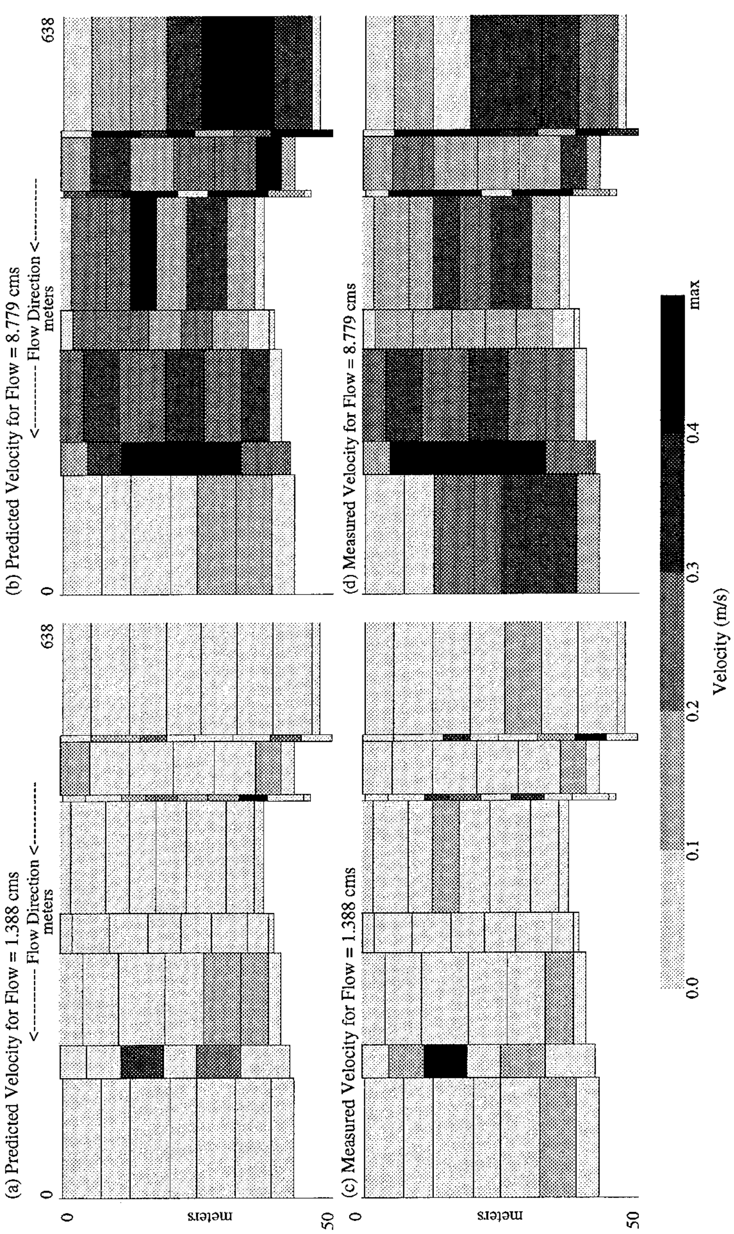


Figure 1. Comparison of predicted and measured velocities for each cell of our representative reach.

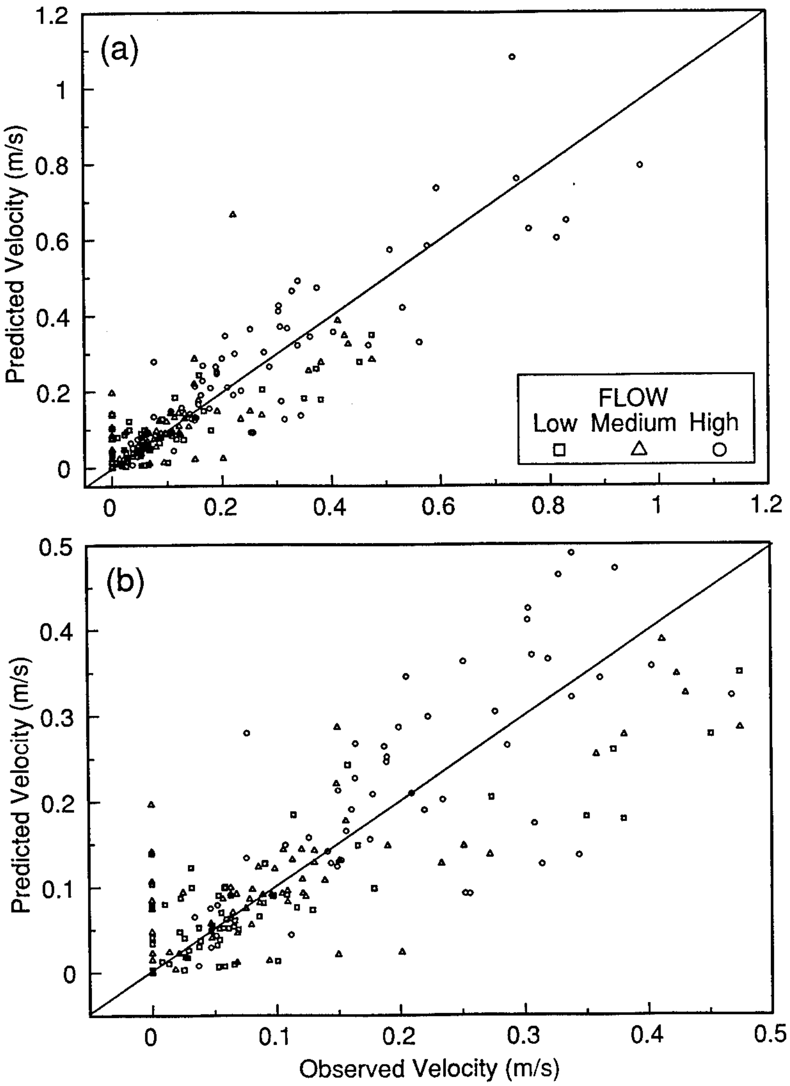


Figure 2. A comparison of predicted and measured velocities based on a PHABSIM survey and analysis of a representative reach of the North Anna River, Virginia. The comparison is made for three flows: 1.4 m^3/s, 2.8 m^3/s, and 8.8 m^3/s for (a) the full range of measured velocities and (b) excluding measured velocities greater than 0.5 m/s.

the complete range of measured velocities. We are mainly interested in predicting the velocities accurately over the range shown in Figure 2b, since even velocities of 0.2 m/s are considered extreme enough to destroy smallmouth bass nests. Regression analysis over the full range of measured velocities shows reasonable agreement between the predicted and observed velocities (r^2 = 0.73, predicted = 0.83· observed + 0.03). In general, the predictions improve at higher calibration flows, with slopes increasing toward the expected slope of 1.0. For the low flow, the slope is 0.47 (r^2 = 0.30); for the middle flow, the slope is 0.68 (r^2 = 0.51), and for the high flow, the slope is 0.88 (r^2 = 0.79).

These results help us to define the appropriate spatial scale of our predictions and guide us in the use of velocity as a habitat variable in the model. It is clear from these results that the modeling capabilities afforded by PHABSIM, combined with the expense of collecting field measurements with a suitably small cell size, may place serious limitations on our ability to predict effects of flow on a small spatial scale. This suggests, for example, that we can expect to have trouble predicting the velocities to which nests in particular sites are exposed. Research is needed to identify alternative hydraulic simulation techniques with modest requirements for field data that can be used at this scale.

In the absence of these improvements, we must restrict our modeling goal to predicting the effect of flow on smallmouth bass in a theoretical reach that shares the statistical habitat properties of our representative reach. We can treat cell depths and velocities in the model as stochastic values generated from a distribution with the PHABSIM predictions as an average. The simulated variation around the cell average can be designed to mimic the differences in the average velocity experienced by individual fish within a cell, or even between multiple cells visited on a given day. In this context, model predictions of fish growth, survival, and reproduction apply to the representative reach and not to individual cells. An important step in verifying the predictions is to ensure that the velocity and depth predictions for the reach are not biased and that they provide a reasonable representation of habitat conditions available to fish in the reach under different flow conditions.

Simulation of Fish Growth and Survival. The flows during this year were low and stable compared with other years. Habitat suitability criteria were developed in this river for young-of-year smallmouth bass (Sabo, in preparation), juvenile smallmouth bass (Groshens, 1992), and for spawning (Joe Lukas, Virginia Polytechnic Institute, personal communication, 1992). The weighted usable area for spawning, larval and juvenile smallmouth bass in this representative reach of the North Anna River in Virginia fluctuates over the growing season in response to flow (Figure 3). In contrast, the numbers of larval and juvenile smallmouth bass predicted by our simulation model increase as a result of reproduction in late spring and early summer, and then decline as a result of attrition through the rest of the model growing season.

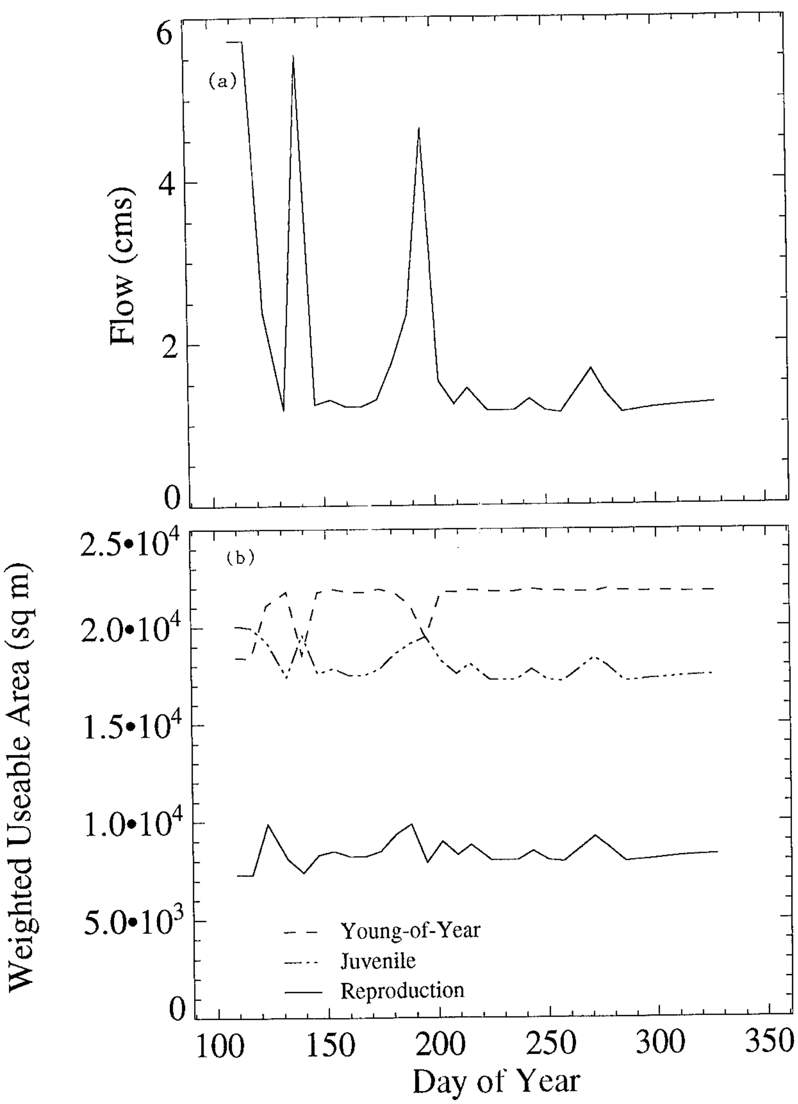


Figure 3. Seasonal patterns in 1991 for (a) measured stream flow and (b) weighted usable area available for three lifestages (spawning adults, fry, and juveniles < 200 mm in total length) of smallmouth bass in a representative reach of the North Anna River in Virginia.

Simulation of Fish Movement. Dispersal of juvenile smallmouth bass from the nests in the model occurs at roughly the same time as dispersal is observed in the field — 15 to 20 days after larvae reach the swimup stage. Model fish are dispersed throughout the representative reach by the time the last juveniles leave the nest, approximately 30 days after nesting commences.

Habitat utilization by model juveniles can be visualized by comparing maps of the representative reach for velocity, depth, and the numbers of fish in each cell. Figure 4 shows maps of this type for the peak flow on a simulation date in mid July (see Figure 3a, day 195). The population on the day shown by the map consists of 390 model fish that range in size from 55.7 to 88.6 mm. On this particular day, the highest density occurs in a moderately deep riffle cell. Figure 4 is presented here to illustrate the types of model results and visualization capabilities available.

CONCLUSIONS

We determined that our model predictions for individual cells would benefit from improvements in the hydraulic simulation component and welcome suggestions for improving velocity prediction at a small spatial scale. Reach-scale predictions can be improved by enhancing our ability to generate theoretical stream habitats with realistic statistical properties and physical constraints. The basic problem is to develop a statistical model to allocate flow among the cells in a transect with the predictions conditioned on local information (e.g., cell depth, Manning's n estimates at each flow). In addition, we anticipate a need for a statistical model to generate realistic theoretical streams without conditioning on local information in situations where the cost of collecting those data is not justified.

We conclude that our model shows promise for predicting the relationship between alternative flow regimes and smallmouth bass recruitment. We have plans to validate various aspects of model predictions including the habitat utilization of young-of-year smallmouth bass, the timing of nest building activity, the selection of nest locations, and their ultimate success.

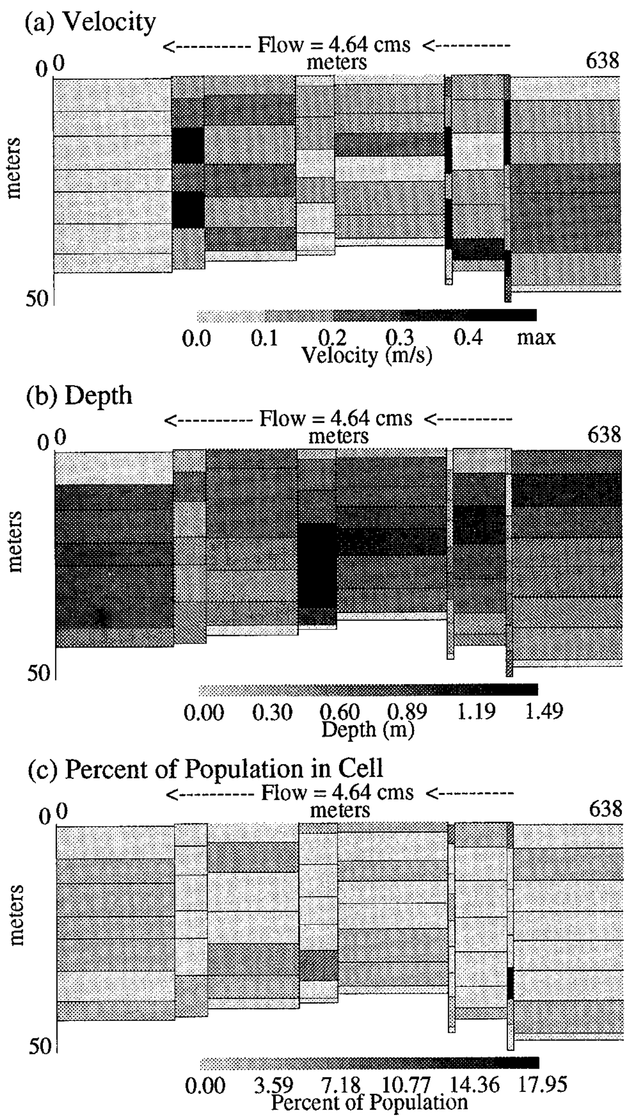


Figure 4. Simulated (a) velocity, (b) depth, and (c) the percentage of the smallmouth bass in each model cell of the representative reach on July 14th.

REFERENCES

DeAngelis, D.L., L. Godbout, and B.J. Shuter. 1991. An individual-based approach to predicting density-dependent dynamics in smallmouth bass populations. *Ecological Modelling* 57: 91-115.

Groshens, T.P. and D.J. Orth. 1992. Progress report on: Assessment of the transferability of habitat suitability criteria for smallmouth bass, *Micropterus dolomieu*. July 3, 1992.

Jager, H. I., D. L. DeAngelis, M. J. Sale, W. VanWinkle, D. D. Schmoyer, M. J. Sabo, D. J. Orth, and J. A. Lukas. An individual-based model for smallmouth bass reproduction and young-of-year dynamics in streams. submitted to *Rivers*.

Sabo, M.J. *Growth and performance of age-0 smallmouth bass in the North Anna River, Virginia*. PhD Dissertation. Department of Fisheries and Wildlife Science, Virginia Polytechnic Institute and State University, Blacksburg, Virginia. in preparation.

Sale, M.J. and R.G. Otto. 1991. Improving the assessment of instream flow needs for fish populations. *Waterpower '91: A New View of Hydro Resources*, Vol. 1:76-84, The American Society of Civil Engineers, New York.

Flow Computation and Energy Performance Analysis For Hydropower Components

Charles C. S. Song[1], Xiang Ying Chen[2] and Jianming He[3]

For the past decades, Computational Fluid Dynamics (CFD) method is increasingly applied to many fluids engineering practices. It is believed that the CFD method will become an effective and less costly tool for engineering design of complex flow configurations. In hydropower engineering, however, the fluid component design is still mainly based on model experimental test and previous experience partially due to the complicated flow configuration although some computational results of flow through turbine runner and draft tube have been reported by a number of investigators (Vu and Shyy, 1990; Song et al., 1991).

In the recent years, the authors have developed a large eddy simulation approach based on three–dimensional compressible hydrodynamic Navier–Stokes equations and applied it to various flow problems, including flow through the three major fluid components in a hydropower unit: penstock bifurcator, turbine runner, and draft tube. This poster will present the computed results of flow through the the three components. The main flow characteristics in the components at given geometry were successfully predicted. Their energy performance was also fairly assessed based on the computed results. In addition, by taking the advantage of the detailed flow information provided by the computation, it was found that the computational model can be used to optimize the geometry design by modifying the original parameters to obtain the best energy efficiency. More detailed description of the computational model is given by Song et al. (1993).

---

[1]Professor, [2]Research Assistant, [3]Research Associate, *St. Anthony Falls Hydraulic Laboratory, Dept. of Civil and Mineral Engineering, University of Minnesota, Minneapolis, Minnesota* 55414, *U.S.A.*

1. Penstock Bifurcator:

Penstock bifurcator is widely used in a multi–unit hydroelectric power plant. Energy loss due to flow separation and vortex shedding at junction plays an important role in determining the energy efficiency of the power plant. The vortex loss, commonly called minor loss, is strongly associated with the geometry of the bifurcator. It is very difficult to obtain detailed flow information using current experimental techniques. It is also difficult to extrapolate the model testing results to the corresponding prototype condition due to scale effect.

The poster shows three–dimensional color pictures of the computational results of the flow through a bifurcation pipe system of a pumped storage hydroelectric plant. Both the generating and pumping modes were tested. The effect of the sickle plate installed at the junction was also assessed. In addition, the computational model was directly used to simulate the prototype condition by increasing the Reynolds number.

2. Turbine Runner:

Accurate prediction of flow within a turbine runner is extremely important for further improvement of a hydropower efficiency since it usually takes the largest part of the total energy loss in a hydro power unit. On the other hand, the flow through a turbine runner is highly turbulent three–dimensional flow, which has very complex flow structure.

This poster uses an advanced three–dimensional graphic technique on an IRIS workstaion to display the complicated three–dimensional configuration. The computed results were just furnished for STS HydroPower Ltd (Song and Chen, 1992). The francis runner was designed with the aid of the computational model to seek for the best energy performance. The computed cases include the best efficiency condition and the maximum gate opening condition. The poster presents the detailed flow patterns which are strongly associated with the runner efficiency.

3. Draft Tube

The draft tube of a hydraulic turbine unit has a very complicated geometry. Its main objective is to recover the remaining kinetic energy from the turbine runner by converting the kinetic energy into pressure head as much as possible. The draft tube must operate under different turbine load conditions, which produce different inflow pattern to the draft tube.

The poster displays the computational results of flow through an elbow draft tube with a dividing wall on symmetrical mid–plane under two different inflow conditions: uniform inflow without swirl and non–uniform flow with swirl. The latter case is based on the output of the runner flow calculation at the runner exit.

Concluding Remarks

The computational results were compared with available experimental

data. It has been found the computational model code is capable of predicting well the flow characteristics in various fluid components of a hydropower unit, and can be used as a general tool for assisting the design and development of a high efficiency fluid component.

Acknowledgment

This work has been supported by the Legislative Commission of Minnesota Resources and the Minnesota Supercomputer Institute of the University of Minnesota, Sinotech Foundation for Research and development of Engineering Science and Technologies, and STS HydroPower Ltd. The authors are grateful to these supporters.

References

Song, C. C. S., He, J., and Chen X. Y., 1993, "Three–dimensional numerical analysis of head loss in a bifurcation pipe flow," to be presented at *An International Conference on Hydropower, Waterpower'93*, Nashville, TN, August 10–13.

Song, C. C. S., and Chen X. Y., 1992, "Mathematical modeling of STS series 60 turbine," St. Anthony Falls Hydraulic Laboratory, University of Minnesota, Research Report No. 331.

Song, C. C. S., He, J., and Chen X. Y., 1991, "Computation of steady and unsteady 3–dimensional flow through francis turbine and draft tube," *1991 International Joint Power Generation Conference, ASME*, 91–JPGC–Pwr–25, San Diego, California, Oct. 6–10.

Vu, T. C., and Shyy, W., 1990, "Viscous Flow Analysis as a Design Tool for Hydraulic Turbine Components," *Journal of Fluids Engineering*, March, Vol. 112, pp. 5–11.

US Army Corps of Engineers
Hydropower Operations and Maintenance Management Innovations

Craig L. Chapman, PE[1]

Abstract This paper tells who the US Army Corps of Engineers is; what our role is in the power industry; what the present and future challenges are and how we are meeting them. The innovations and initiatives that are discussed include performance indicators, equipment condition indices, and risk analysis.

Introduction The US Army Corps of Engineers operates and maintains 75 power projects with 349 main units. This 21,000+ MW capacity represents approximately one quarter of the US hydropower capacity or approximately 3% of all capacity. When compared to the nation's electrical utilities, the Corps of Engineers is the fourth largest and the single largest hydropower operator.

With power projects all across the US, we are presented with more diverse issues than most electric suppliers who are more regional. We have the challenges and opportunities that come with multi-purpose projects. Impacts on navigation, flood control, recreation and the environment must be considered in all of our hydropower activities (and visa versa).

Corps of Engineers water resources projects are constructed, operated and maintained with funds appropriated by Congress. For hydropower, the Power Marketing Administrations (Southeastern, Southwestern,

----

[1]Hydropower Coordinator, Operations Branch, Headquarters, US Army Corps of Engineers, 20 Massachusetts Ave., NW Washington, DC 20314-1000

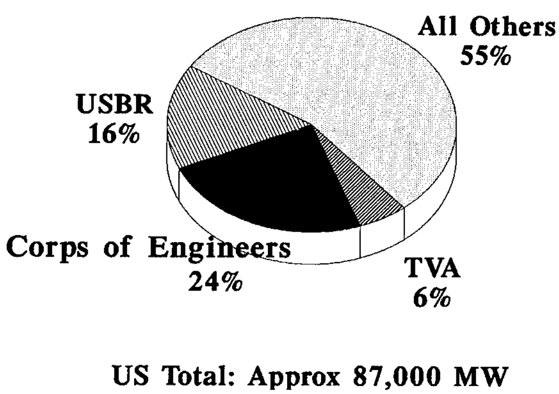


Figure 1. Hydropower Capacity in the US

Western Area and Bonneville) set their power rates to recover the cost of the capital investments and routine O&M. Approximately $400-500 million is paid back to the US Treasury each year on an average generation of 80 billion kW-Hrs.

The Corps of Engineers prides itself on its water resources projects. No matter how hard one tries, nothing lasts forever. While we do not replace equipment based on age, the older the plant the more "opportunities" we have for major repairs or replacements. Nearly 40% (133 of 349) of our units are 35 years old or older. This puts many of our power plants near the end of the design life for major components such as switchgear, turbines and generator high voltage windings. The time to replace equipment is when it is at the end of its economic life. That point is when, regardless of calendar age, it is no longer practical to continue maintenance.

Budget caps have increased competition within the organization for funding and manpower. This presents the operations and maintenance (O&M) management, from the power plant superintendent to the Chief of Engineers, with new and increasingly difficult challenges. Funding for hydropower O&M had been essentially level through Fiscal 1990 (FY90). For FY91 and FY92 an added emphasis was placed on hydropower maintenance, especially the non-recurring major items. This was not extra funding, it was a re-prioritization of funds within the Corps of Engineers budget ceiling. This was especially difficult at a time when the Federal budgets were and still are under tremendous pressure.

As we began to look at what would be required to meet the future needs of our projects, several initiatives were identified and put into place. These initiatives have application in most all O&M activities: flood control, navigation, construction, etc., in addition to hydropower. These initiatives are: performance indicators, equipment condition indices, risk analysis and an investment strategy. These will be discussed individually.

Hydropower Performance Indicators: Within the Corps of Engineers operations community, performance indicators have been used since FY87 as a "report card" to compare in broad terms our water resources projects. The performance indicators that are reported for hydropower are: plant availability, forced outages per unit, manpower per unit, maintenance costs per unit and operating costs per unit. The availability and forced outage rates are reported quarterly, while the remainder are reported on an annual basis. Unlike some activities, most of the data used for the indicators was already being collected for use in reports to headquarters and/or the FERC's Energy Information Agency.

The hydropower managers are finding the performance indicators useful. This is especially true now that we have five or more years of data for all plants. By comparing their projects' data from year to year and with other projects, a manager can get a sense of how they are doing. There have been many instances where the performance indicators have pointed out areas that needed management's attention.

The manpower and O&M costs per unit do not have goals assigned to them, the yearly trends are the most important. The goal for forced outages per unit (less than 1.4/year) is being met by most Districts and Divisions the majority of the time. The unit availability goal of greater than 93% has not been met Corps-wide in the last three years due to outages for major planned modifications, in addition to reliability problems.

Our goals and our actual experience are as good or, in many cases better, than the hydropower industry as a whole. This office just concluded a series of meetings on performance indicators with the four Power Marketing Administrations (PMAs). The meetings were to show the PMAs what data we were collecting and to solicit input from them as to what was useful and what other information would be of value. One of the items that we came away with was the need to report our unit availability so that there is a distinction between planned and unplanned outage time. There also is interest in reporting maintenance cost per

unit by routine and non-routine categories. We are also looking at the Generation Availability Data System (GADS) and the data being collected by the Electric Utility Cost Group (EUCG) in an attempt to collect and present our data in a form that is usable to the industry.

<u>**Hydropower Equipment Condition Indices**</u>: Anytime that the condition of equipment is discussed (budget requests, design documents, etc.) the question comes around to "how good or how bad is it". Equipment condition indices are a way to quantify the question.

Within the Corps' water resources activities, equipment condition indices began in the navigation lock arena. Attempting to quantify the condition of a lock wall monolith can be a very subjective task. The process to quantify the condition of equipment began as a part of the Corps' Repair, Evaluation, Maintenance and Rehabilitation Research Program (REMR). REMR Technical Note OM-XX-2, March 1990, defines the concept of a Condition Index.

The Hydroelectric Design Center (HDC) in Portland, OR, was tasked with developing the condition index procedures for hydropower. The approach used for developing the indices was to use the results from accepted test procedures and convert these values into a condition index of 0-100. This approach provides good repeatability and a sound technical basis. The draft document was distributed for field review in early 1991. A field review group meeting was held in September 1991. From that meeting several of the Districts and Divisions took on the task of field testing the condition index procedures for various major hydropower equipment. The results of this exercise was used to calibrate the procedures. The latest version of the condition indices is being prepared for printing and distribution in early 1993. There is still more work to be done in this area. Development of procedures for auxiliary equipment and refinements of exiting major equipment will make this an ongoing process.

<u>**Hydropower Risk Analysis**</u>: In the mid-1970's the major rehabilitation program was initiated to provide a way to manage the large expenditure of funds required to extend the life of our aging projects. Over the years the program and procedures have undergone many changes. With the Water Resources Development Act of 1986 came the establishment of the Inland Waterways Fund and cost sharing of inland navigation rehabilitation. Better evaluation reports were needed.

Guidance has been prepared that added the concept of risk analysis to the evaluation report. When a project is in need of rehabilitation, total replacement is not always the correct answer. In the preparation of an evaluation report all practical alternatives are investigated, including not doing anything other than continued maintenance and repair. In addition to the engineering solutions, each alternative's environmental, economic and cost are evaluated. Refinements in the program continue as we improve our risk analysis techniques.

The condition of a structure or a piece of equipment and how it is operated and maintained all are evaluated. This is one place that the condition indices and performance indicators come into play. Probabilities of unsatisfactory operation (or failure) are developed for the project. Structures use a methodology that looks at design capabilities, existing capabilities and expected loads to determine the probability of unsatisfactory operation.

For electrical and mechanical equipment, especially in hydropower, the design life and condition index are used to develop the probabilities. We use survivor curves that give the expected percentage of equipment that will be in service after a period of years. These are similar the mortality tables that the life insurance companies use to set policy rates. The expected failure rate at any given time is influenced by a function of the condition index. The better the condition, the lower the risk; the poorer condition, the higher the risk. The risk is the slope of the curve. By using simulation techniques, costs over time can be developed to complete the economic analysis.

Hydropower Investment Strategies Study: In order to plan for the future it became apparent that we needed a long range plan to maintain our facilities. In early 1991, the Corps' Institute for Water Resources (IWR) at Ft. Belvoir, VA, was tasked with developing a plan of study for an investment strategy that would be used as a business plan for the future. A taskforce was formed with representation from Engineering, Operations, and Planning at both the field and headquarters level, plus the IWR staff. The taskforce has met several times during 1991 and 1992. The study presents seven recommendations: (1) develop a ten year plan (investment needs inventory); (2) develop an inventory of uprating potentials; (3) share information with the hydropower industry; (4) continue the work in economic and risk analysis; (5) develop ranking procedures for investment needs; (6) promote upfront financing; and (7) establish a Corps/PMA taskforce to develop additional management issues (future). The final draft of the study is being edited and will be completed in the near future.

A second working group has begun the investments needs inventory and the development of the ranking procedures, recommendations (1) and (5), respectively. The raw data for FY1993 through FY2002 has been collected. The ranking procedures are being established to allow prioritization of the field data. The data call has been made to begin the uprating inventory for recommendation (2).

Summary: This is a brief description of the major Corps-wide activities related to management of these renewable resources. If you have questions, comments, or would like to share information on similar innovations, please call or write the office listed in this paper.

Dam Modification For Recreational Boating

Richard E. McLaughlin[1]

Abstract

White water recreational facilities associated with hydropower relicensing are usually confined to those facilities that allow white water enthusiasts to portage around dams, warning signs, and provisions that regulate release rates to allow for downstream boating. To the author's knowledge, facilities that allow white water craft passage (white water bypass structures) around dams have not been included in any constructed hydropower project. They have, however, been successfully designed, modeled, constructed, and operated on other types of low-head dams in the U.S. These structures can be designed to simultaneously serve as fish ladders, to be part of park amenities, and to enhance the recreational value of waterways and environs. Design-related issues are discussed and a brief review of representative projects is included.

Introduction

The increased recreational use of rivers by boaters and the safety concerns associated with dangerous hydraulic currents that occur in low-head dams and drop structures have led to development of white water bypass structures (boat chutes). These passages offer kayakers and rafters a route around a dam that is otherwise a barrier to white water craft. The first white water bypass (Confluence Park in Denver, Colorado) was constructed in 1976. Since then, other bypasses have been designed and retrofitted to existing dams. These multi-purpose structures have confirmed that, if correctly designed, they are technically and economically feasible and offer a number of recreational, safety, and environmental benefits. In spite of their benefits, to the author's knowledge white water bypasses have only been included in one Application for License to the Federal Energy Regulatory Commission, and none have been constructed as part of a hydroelectric project.

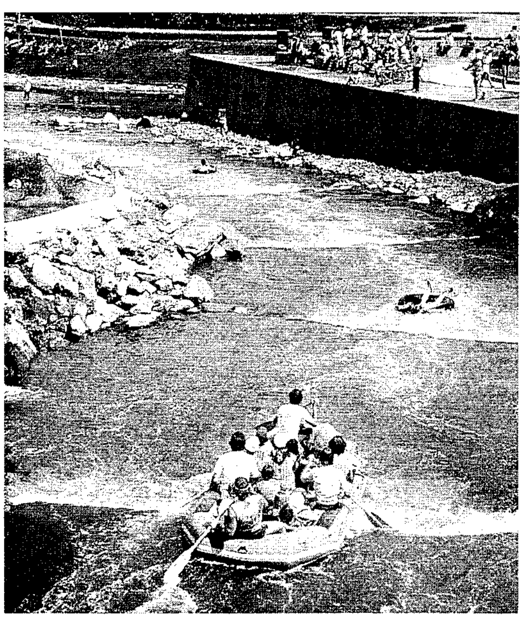
---

[1]Principal, McLaughlin Water Engineers, Ltd., 2420 Alcott Street, Denver, CO 80211

Bypass structure design has greatly advanced in the last 17 years. Experience with constructed projects and with large-scale physical model studies has increased available knowledge about the complex currents that occur in the bypasses and has provided data and experience concerning critical design elements. This increased knowledge has led to more predictable performance and lower construction costs. The author's experience with projects he and McLaughlin Water Engineers, Ltd. designed is discussed below, illustrating the benefits and key design elements of white water bypass structures.

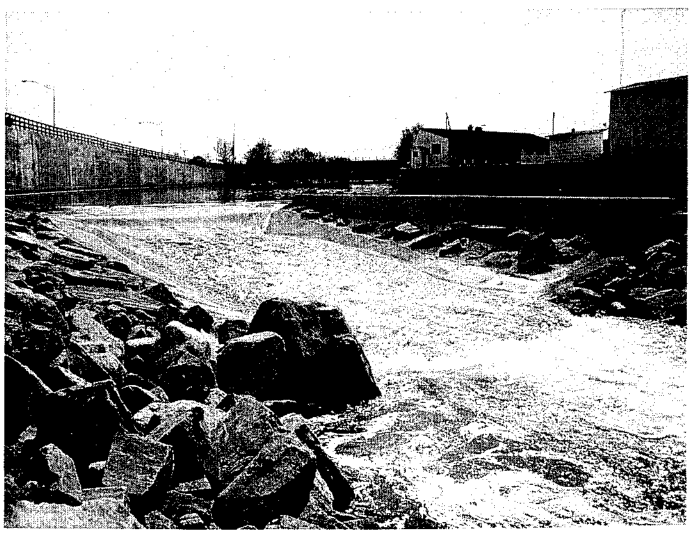
Benefits

Recreation. The chief benefit of including a white water bypass around a dam is, of course, the enhanced recreational value to river enthusiasts. Portages detract from the white water experience, but white water bypasses can be exciting because of the drop created by the dam. White water bypasses can offer a wide range of recreational opportunities. The Confluence Park bypass was designed and built as a focal point of the interconnected riverside parks and pathways in the Denver metropolitan area.



Boat chute at Confluence Park near downtown Denver.

Currently, this park is being further upgraded with a new chute configuration, park features, riverside paths, and lighting. Alternatively, projects such as the bypass at Third Avenue in Denver have been built to meet strict budget and space requirements. One chute/pool at Third Avenue drops about five feet over a 130-foot length.



Boat Chute at Third Avenue in Denver.

Safety. Safety is an important benefit of white water bypasses. Although the safety of bypass structures has elicited concern from various individuals, the increased safety that they provide has been a driving force in implementing several bypass retrofit projects, primarily because of the dangerous currents that are created by conventional dams and drop structures.

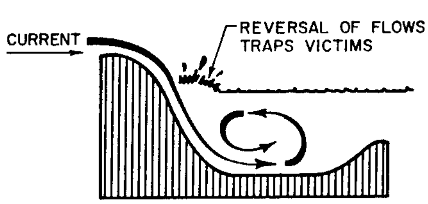


Figure 1. Dangerous Currents

Although warning signs are typically placed upstream of dams, children, fishermen, and boaters have been trapped in these sometimes tranquil-looking currents. Many people have drowned at dams throughout the U.S. and Canada (Makuk, 1988). At the Third Avenue Dam, and other dams where multiple drownings have occurred, no drownings have been reported after retrofitting the dam with a bypass.

River Ecology. Improving river ecology and lowering the floodplain can also be beneficial results of adding a white water bypass. Because a bypass is essentially a hydraulic conveyance structure, it can lower the floodplain upstream of a dam. All of the bypass structures designed by MWE have lowered the existing regulatory floodplain. Environmental benefits such as reaeration can improve water quality downstream of the bypass. A current study of the reaeration created by bypass structures will help to quantify their impact on water quality.

Fish ladders can be incorporated into a white water bypass structure. A white water bypass at Horseshoe Bend on the Payette River in Idaho was the first bypass designed (but not yet constructed) with a fish ladder. In addition to white water design concerns, the bypass was designed to create the velocities and flow configurations necessary to attract several varieties of fish. By combining a white water bypass and fish ladder, capital costs are reduced and flow bypass requirements are minimized.

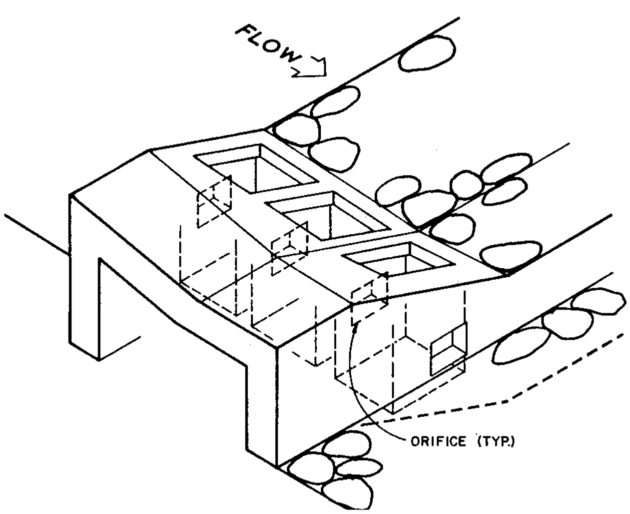


Figure 2. Fish Ladder Incorporated into Boat Chute Design

Elements of Design

Although much has been learned in designing 12 bypass structures and associated facilities, each site and each dam present unique challenges. Because of high Reynolds numbers, air entrainment, and the three-dimensional flow involved in a white water bypass, mathematical techniques and one-dimensional hydraulic analysis cannot be directly or reliably applied. Large-scale model studies have been conducted to refine site-specific designs and to determine semi-empirical design methods. These models have been critical to developing bypasses that can efficiently dissipate energy while creating currents and hydraulic jumps that do not trap boaters. Designing for boating safety is especially challenging because of the wide variations in flow and tailwater that the facilities incur. For boating safety, experience has demonstrated that a series of chutes and pools is preferable to one large drop. The pools allow for recovery from mishaps, and the small drops are easier to navigate. Multiple chutes and pools can also accommodate a variety of drop sizes and site requirements. Chute design is crucial to the formation of the hydraulic jumps, waves, and eddies that occur in the downstream pool. The author has reported previously (McLaughlin, Grenier, 1990) on chute design and related hydraulics.

In addition to boating and hydraulic issues, there are site parameters, such as dam height, low-flow conditions, flood flows, boating season, location of the intake works, topography, sediment loading, and fish migration, that are unique to each dam and site. These parameters affect the number and configuration of the chutes, the length of the pools, drop height, pool depth, alignment, channel armoring, and grading. Pool length is a key design factor; pools must be long enough to dissipate hydraulic energy and allow recovery time for boaters, yet longer pool lengths result in higher capital costs. The drop in each chute, depth of pool, and design flow rates are all interdependent with pool length. These factors are also used to determine the number of chutes and pools needed.

Construction materials can include reinforced concrete, steel, aluminum, riprap, grouted rock, boulders, geotextiles, and plastics. The materials used depends on their availability, the project budget, maintenance requirements, low-flow conditions, and the desired aesthetics. Highly variable and powerful flows require materials that can withstand the resulting velocities and forces; the placement and thickness of materials must be appropriate for the velocities and forces, which are predicted.

The entrance to a white water bypass is also an important design feature. Signs, warning buoys, guide buoys, and river markers must be considered in directing boaters to the entrance. The location of the entrance is dependent on sediment concerns, curvature of the river, and location of the intake structure. Because of the desire to limit or manage bypassed flows, the crest of the entrance is usually an important design issue. A movable crest is sometimes advantageous in regulating flows to the bypass. Several designs for a movable (yet boatable) crest have been developed. While these will increase capital costs and require some maintenance, they are often cost-effective overall.

Conclusion

Based on physical model studies and 17 years of experience, white water bypasses have been shown to be technically and, in many cases, economically feasible. Site-specific investigations of white water facilities on low-head hydroelectric dams may be appropriate for rivers that support white water sports. White water bypass facilities also offer a number of environmental benefits, such as fish passage, aeration, and a reduction in regulatory floodplain elevations. They can be built as part of a multi-use park, for white water competition, or for projects with budget constraints and limited space.

Appendix

Makuk, John S. Multiple Use of Water Resources: Adapting Weirs for Recreation. University of Calgary, Environmental Science Program, January 1988.

McLaughlin, Richard E., Grenier, Roger R. Low-Head Drop Structure Hazards: Modeling of an Abrupt Drop Boat Chute. Hydraulics Division of the American Society of Civil Engineers, Proceedings of the 1990 National Conference, Volume 1, July 20 - August 3, 1990.

Plant and System Benefits Associated with Adjustable Speed Hydro

Peter Donalek, Senior Member Institute of Electrical and Electronics Engineers (1)

Abstract

Plant and system operating benefits of adjustable speed hydro machines are discussed. Constraints such as cavitation, vibrations, oscillations, and efficiency establish power output limits of single speed units in plant operations. Adjustable speed machines can operate through out a range of head and flow rather than at fixed points on a single speed curve. If operating speed is adjustable, then improved pump - turbine efficiency, improved stability, and other associated plant benefits will be realized.

Introduction

It is a well known fact among engineers involved in hydro electric plant design and operation that if the speed of the generator/ motor could be adjusted as an operating variable, then improved turbine efficiency and operation at greater head variations can be realized.

Until recently, adjustable speed control for large size generators and motors has not been practicable for commercial application in hydro electric plants. Developments in power electronics have made available robust and reliable high ampacity thyristor devices along with necessary control systems. With the availability of adjustable speed machines it is now possible for hydro

----

(1) Harza Engineering Company, Electric Power Systems Department, 233 S. Wacker Dr., Chicago, IL 60606.

plant planners and designers to reconsider basic design limits.

Electric machine designers in Japan have restudied adjustable speed machines, and have made installations in pumped storage and conventional hydro projects. Three adjustable speed hydro projects have been reported in the engineering literature. One is the 22 MW test installation at the Narude plant. A second project is the 87.4 MW replacement machine at the Yagisawa plant.

Based on the experience gained from these two projects, the Kansai Electric Power company is installing two 390/320 MW adjustable speed generator-motors in the four unit Ohkawachi pumped storage plant. The speed variation is plus minus 8% This installation appears to have been justified on the basis of the frequency regulation capability in both pumping and generating modes.

Adjustable Speed Technology

The adjustable speed machine has a laminated rotor with a smooth three phase winding, and is otherwise similar to a conventional salient pole synchronous machine. The rotor is excited by an adjustable frequency excitation system. The frequency of the stator field is at the 60 Hz synchronous frequency of the connected utility system.

The rotor shaft speed is equal to the algebraic sum of the synchronous frequency of the stator and the rotor frequency. Thus shaft speed is adjusted above or below the synchronous speed by changing rotor excitation frequency. Since there are two electrical connections one for the rotor and one for the stator the machine is called "doubly fed"

Potential Benefits

A partial list of potential primary benefits that should be achievable with adjustable speed hydro includes the following:

Increased Efficiency. Adjustable speed machines will allow mechanical speed to be set at the pump-turbine maximum efficiency point at all times. Hydro plant operators will be able to fine tune turbine efficiency and track variations in head as reservoir levels or available flow changes. In pumped storage plants, adjustable speed

machines will allow operators to adjust for optimum efficiency in both the pumping and generating modes.

Adjustable Pumping Load. Because machine speed can be adjusted in the pumping mode, system dispatchers can set pumping load to match system power flow limitations and amounts. With this added dimension, system dispatchers can extend pumping time and match pumping load with available energy.

Pump Mode Frequency Regulation. With adjustable speed machines it is possible to provide frequency regulation in the pumping mode. Units at the Ohkawachi project have nominal ratings of 320 MW generating and 350 MW pumping. With adjustable speed capability the units have a power range of plus minus 102 MW as generators, and plus minus 80 MW in the pumping mode. The adjustable capacity range allows the units to be used for load frequency regulation in pumping mode as well as generating mode.

Improved Load Tracking Capability. Adjustable speed machines have a broader operating range in the generating mode as compared to single speed machines. The broader operating range together with fast active power response gives this class of machine a better load tracking capability and decreases frequency fluctuations on the bulk power system.

Operation at Reduced Flow. Adjustable speed hydro allows operation over an increased range of flow conditions. Many plants are being required to operate under water availability and regulation conditions that are significantly different from those for which the plant was originally designed. With adjustable speed machines it will be possible to operate plants efficiently under revised flow and head conditions.

System Dynamic Performance. High speed transient stability support is possible because adjustable speed machines can interchange (supply or absorb) discrete amounts of energy between the machine rotating inertia and the power transmission system.

Increased Transmission Capability. Adjustable speed machines can respond, to changes in the power flow levels on the transmission system in a few electrical cycles. Thus it will be possible to operate transmission lines closer to their steady state stability limit, and will allow transmission line capacity to be more fully utilized.

Potential Environmental Benefits. Studies have shown that survival rates for fish passing through turbines is greatest when turbines operate at maximum efficiency. With adjustable speed hydro units it should be possible to operate at maximum efficiency at critical fish travel times, thus increasing fish survival rates.

Potential For Rehabilitation and Upgrading. Based on economics and specific site conditions, the above benefits can be realized as part of plant upgrade and rehabilitation programs.

Miscellaneous Benefits

In addition to the primary benefits there is a set of miscellaneous benefits that may also be considered. These include:

Dual control capability in which machine speed can be controlled either by governor action or by excitation frequency.

Partial load operation improvement resulting from ability to change operating point

Shoulder pumping prior to and after full load pumping

Dynamic braking results in reduced use of brakes and possible energy recovery on run down.

Reduced throttling due to ability to change operating point

Synchronous condenser operation at low speed means reduced windage losses

Staring duty in pump mode using Cycloconverter.

Conclusion

There is a need in the industry for technical - economic analytical models of adjustable speed hydro plants. The models would allow engineers and managers to evaluate the benefits and performance of conventional and pumped storage hydro plants with adjustable speed machines.

Investigation on Low-Cost Flowthrough Rockfill Dams and Spillways - A Summary

T. P. Tung,[1] V.K. Garga,[2] and D. Hansen[3]

Abstract

The hydraulic behaviour and stability characteristics of flowthrough rockfill dams was investigated in this study. The experimental investigations consisted of one dimensional flow studies in a packed column, and of two dimensional flow studies on model dams in three flumes, including a flume study at the low temperature laboratory of the National Research Council.

An investigation was made of the choice of non-Darcy flow equation, and on the estimation of the empirical parameters in such an equation. The overall hydraulic behaviour of flowthrough rockfill dams was characterized, including development of stage discharge curves and computation of the phreatic surface. Modifications to methods appearing in the literature are proposed. The stability of the downstream slope of flow through rockfill dams was investigated, including physical measurements of the force needed to prevent failure. A means of modelling non-Darcy pore pressures is presented. It was found that when non-Darcy pore pressures were used for rotational failure analyses the

[1] Mr. Tony Tung, Manager, Hydraulic Energy Program, Efficiency and Alternative Energy Technology Branch, Department of Energy, Mines and Resources, 580 Booth Street, Ottawa, Ontario, Canada, K1A 0E4, (613) 996-6119

[2] Dr. Vinod Garga, Professor and Chairman, Department of Civil Engineering, University of Ottawa, Ontario, Canada, K1N 6N5.

[3] Dr. David Hansen, Assistant Professor, Department of Civil Engineering, Memorial University, St. John's, Newfoundland, Canada.

results of such analyses appear to be very conservative. It is suggested that the mechanism of failure assumed in methods such as Bishop's Simplified Method, namely, rotational failure along a limit-equilibrium slip circle, is often invalid for flowthrough rockfill dams. A new method of analyzing the stability of flowthrough rockfill dams is presented. This method emphasizes the detailed consideration of seepage forces within two adjacent wedges.

A study of a model flowthrough embankment in the low temperature laboratory of the National Research Council of Canada, combined with a theoretical examination of the heat flux budget for flowthrough rockfill dams, indicated that flowthrough rockfill dams are not usually prone to significant changes in hydraulic capacity due to ice accretion.

Introduction

Flowthrough rockfill dams and spillways are perceived to be an attractive alternative for the design of low-head hydro schemes because they may reduce the costs of total civil works, particularly in the remote northern climate applications. A low-head rockfill dam with a built-in "flow-through" section, capable of handling the design return period flood event, passing some water to maintain minimum flow, or providing upstream regulation and reducing spill during small flood events, could lead to reduction of storage development costs by reducing or eliminating expensive concrete construction.

The Department of Energy, Mines and Resources of the Canadian government has been supporting the Department of Civil Engineering at the University of Ottawa for experimental and theoretical research on flowthrough rockfill for the last four years. The research, currently the only one of its type in North America, has so far focused on the hydraulic behaviour and stability characteristics of flowthrough rockfill.

Research Aspects

The principal aspects of this research are as follows: (Hansen, 1992)

a) A multi-disciplinary consideration of most of the equations describing non-Darcy flow.

b) An experimental program has been carried out in five areas

 i) particle characterization,
 ii) packed column tests,
 iii) parametric study in glass-walled flume, (figure 1)

iv) additional measurements using large rock in a floor flume, (figure 2)

v) experiments in a low temperature laboratory.

c) A new computational method to estimate the quantity of flow through the rockfill embankment has been developed.

d) A detailed computational framework to enable the prediction of the minimum elevation over which mesh reinforcement is needed to prevent erosion of the downstream force has been developed.

e) A new method to evaluate the possibility of massive deep seated failure has been developed. A less conservative method of estimating need for anchor reinforcement has been proposed.

f) In another innovative series of experiments, the low temperature behaviour was investigated using the Low Temperature Laboratory of the National Research Council of Canada. Based on this research a simple method for estimating the rate of growth of ice at the phreatic surface has been developed.

The University of Ottawa is presently extending the above studies to consideration of the design of overflow rockfill. In many practical cases it may not be feasible to channel the peak flows through the rockfill itself. In such cases, recourse has to be made on the design of an adequate overflow rockfill section. Also, with the use of an upstream concrete facing in many rockfill hydro dams, the option of passing peak flood over the dam becomes particularly attractive.

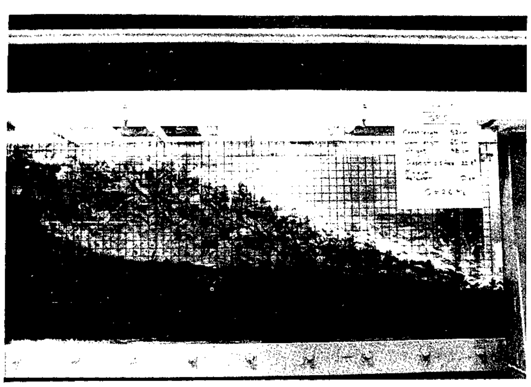


Figure 1: Typical model dam in glass-walled flume

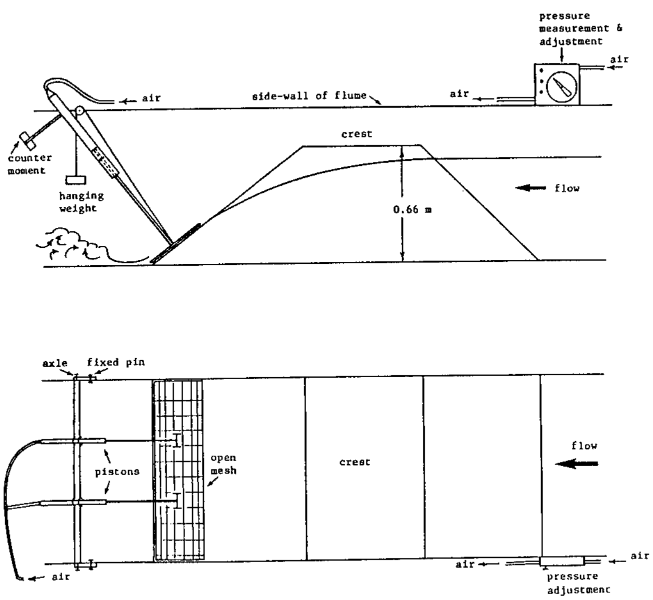


Figure 2. Arrangement of model dam and force-measurement system in large flume.

Summary of Results

The research deals with the hydraulics and stability of "flowthrough" rockfill embankments. This class of embankments is unique in that, unlike ordinary earth embankments, the volume of water moving through the embankment is relatively large. This research provides new information in the following seven areas:

1. A multi-disciplinary consideration of most of the available equations describing one-dimensional non-Darcy flow. Such flow is characterized by a non-linear dependence of velocity on the hydraulic gradient. The nature of this dependence is often expressed in terms of such factors as the porosity of the porous media and the size and shape of the individual particles making up this media. In the context of flowthrough rockfill embankments, the coefficient of variation for the selected empirical parameter (used as an indicator) was about 20%, with this variability decreasing to about 15% for prototype-sized particles. New data is also

presented to assist in the evaluation of the effect of irregular particle shape, as compared to particle spheres.

A method of determining the surface area of crushed rock was developed, and a useful parameter associated with surface area efficiency was described. For non-platy rocks the surface area efficiency ratio, r, was found to be useful in calculating the hydraulic mean radius of rockfill. (Garga et al, 1991)

2. A new computational method to estimate the quantity of flow moving through a rockfill embankment, as a function of the upstream water level, is presented. The general applicability of this method was developed from a parametric study on dams having a variety of configurations. The key equation derived from this parametric study can be used in conjunction with any power-function-form of non-Darcy flow equation.

3. If some minimum upstream water level is required, an impermeable element (in the form of a wall-like hydraulic barrier) will be needed. The hydraulics of an upstream impermeable facing (UIF) was studied in this research program. The hydraulics of the control section through the lip of the UIF exhibited unusual behaviour which could not be described by any single formula. However, the relative efficacy of a range of methods, both new and those presented by previous researchers, is presented and discussed. Some guidance is also presented as to how to account for the effect of an elevated downstream water level on the stage-discharge rating curve of a flowthrough rockfill embankment.

4. Three distinct analytical methods for computing the position of the phreatic line are presented and evaluated. For a given amount of discharge passing through the dam, the phreatic line represents the internal division between the wet and dry portions of the dam. This phreatic line is of interest for two reasons. First, it is needed as a boundary-condition to permit internal pore pressure modelling. Second, it may be used to develop a stage-discharge rating curve, which can be compared with that obtained using the more direct method mentioned in item (2) above. The validity of the Dupuit assumption, which is inherent in all three of the analytic methods, is also discussed. For non-prismatic river channels the engineer must resort to numeric, rather than analytic, expressions to compute the phreatic line, and the general approach for this class of problems is described.

5. A detailed computational framework is presented to make possible the prediction of the minimum elevation over which mesh protection is needed to prevent erosion of the downstream face of the dam. Computational methods are presented for evaluating the

combined destabilizing effects of the spatially-varied overflow and the emergent seepage on individual 'loose' rock particles residing on the downstream face. These methods are relatively general and can be applied to any size dam; however, further research is needed on the role of roughness and on making independent predictions of the depth of water at the toe of the dam.

6. As distinct from erosion of the downstream face (sometimes called unravelling failure), the possibility of massive deep-seated failure was investigated in detail. As a first step to permit such evaluations, non-Darcy pore-pressure modelling was performed. An efficient method for executing such modelling on an electronic spreadsheet is presented. Two types of massive failure were considered. The first, based on classical rotational failure, was found to predict relatively deep-seated failures involving a relatively large fraction of the dam in question. This implied extensive and potentially expensive anchor reinforcement. The second, a wedge-type failure, generally implied mass-movement of considerably less material than did rotational failure. In an innovative series of experiments it was found that the *measured* bursting force at incipient failure was far closer to that theoretically predicted by the wedge method than that predicted by a classical rotational slip-circle method (Bishop's Simplified Method).

7. Most existing prototype-scale flowthrough rockfill dams and spillways have been constructed and operated in countries warmer than Canada, such as Mexico, Venezuela, and Australia. Prior to this research,the behaviour of this type of dam in sub-zero temperatures was therefore a matter of speculation. In another innovative series of experiments, this low-temperature behaviour was investigated using the Low Temperature Laboratory of the National Research Council of Canada. Based on this research a simple method for estimating the rate of growth of ice at the phreatic surface is presented. Ice was *not* observed to form in the voids within the body of the dam, except in one case in a small mixing zone (the base of Parkin's zone 3) Parkin 1966 below the lip of the UIF. A detailed theoretical framework for performing heat budget calculations is also presented, together with a short computer program to facilitate these calculations. It is quantitatively demonstrated that the viscous dissipation of hydraulic energy plays an important role in preventing ice formation in the voids.

References

- Garga, V.K., Townsend, R. and Hansen D. (1991). A Method for Determining the Surface Area of Quarried Rocks. Geotechnical Testing Journal, ASTM, Vol. 14, No. 1, pp. 35-45.

- Hansen D. (1992). The Behaviour of Flowthrough Rockfill Dams. Ph.D.: Thesis, Department of Civil Engineering, University of Ottawa.

- Garga, V.K., Townsend, R. and Hansen, D. (1990). Design of Low-Cost Flowthrough Rockfill Dams and Spillways, Research Report to CANMET, Energy, Mines and Resources Canada.

- Parkin, A.K., Trollope, D.H. and Lawson J.D. (1966). Rockfill Structures Subject to Water Flow. Journal of Soil Mech. and Fdn. Engng. Div., ASCE, Vol. 92, SM6, pp. 135-151.

RAW WATER SYSTEMS EVALUATION FOR
ZEBRA MUSSEL CONTROL

K. Young[1], K.Chen[2], C. Lange[3], T. Short[4]

Abstract

This paper presents the scope and recommendations from a raw water systems evaluation for zebra mussel control conducted at four fossil and four hydro plants operated by TVA.

Scope

Many raw water system components in both fossil and hydro plants are susceptible to zebra mussel infestation. These components may include: embayment walls, stationary trash racks, pumps intake housing, traveling screens, all exposed non-toxic hard surfaces, condenser tubes and tube sheets, heat exchanger water boxes, heat exchanger tubes, strainers, water pipes, valves, storage tanks, raw water service systems, fire protection systems and other systems which use raw water.

[1]Consultant, Water Resources, Ebasco Environmental, 211 Congress Street, Boston, MA 02110-2410.

[2]Senior Engineer, Ebasco Environmental, 145 Technology Park, Norcross, GA 30092.

[3]Senior Scientist, Acres International Corporation, 140 Audubon Parkway, Amherst, NY 14228-1180.

[4]Mechanical Engineer, Acres International Corporation, 140 Audubon Parkway, Amherst, NY 14228-1180.

Ebasco Services Incorporated (Ebasco) was tasked by TVA to perform thermal and hydro raw water system evaluation and to develop the viable control technologies to mitigate the effects of zebra mussels in eight (8) fossil and hydro plants. Assistance in the hydro plant assessments was provided by Acres International Corp.

This study included a raw water system evaluation and recommended treatment options for eight TVA plants - Shawnee, Cumberland, Allen and Johnsonville Fossil Plants as well as Kentucky, Wilson, Pickwick and Wheeler Hydro Plants.

This report documented the following:

1. Identification of systems and plant areas susceptible to zebra mussel infestation.

2. An assessment of the various treatment options available in the industry today.

3. An assessment of the various treatment options for each plant.

4. A recommendation for treatment options for each plant under study.

5. Conceptual design for the recommended system.

6. Cost estimates for the recommended system.

A total of seven categories of promising control technologies were reviewed and thoroughly evaluated for their potential applicability at each of the plants under study. However, only four technologies were considered as proven technologies which have been demonstrated to be effective in controlling zebra mussels in raw water systems. They were anti-fouling coatings, physical/mechanical removal, thermal treatment and chemical treatment.

Recommendations

Chemical treatment was recommended as a principal control option for a short-term eradication to kill the mussels in the infected areas and as a long-term preventive measure to control the settlement and growth of the zebra mussels in all the TVA fossil plants raw water systems studied. Physical/mechanical methods, such as scraping, hydroblasting or heat treatment, were also recommended for use in removing the established mussels in any of the raw water system components such as trashracks which are not protected by the

recommended chemical treatment program.

The recommended chemical treatment system at each fossil plant under study was provided with two chemicals, NaOC1 and NaBr/surfactant and with control flexibility to use chlorination, bromination, or some combination of the two. The chemical treatment system will be designed and equipped with the feed connections, isolation valves, check valves, etc., necessary to allow feeding one or more of the chemicals to all of the major raw water systems. The chemical treatment system would be sized to have adequate capability for continuous treatment to produce up to 0.5 ppm total residual oxidant (TRO) as well as for a periodic "shock treatment" to produce up to 2 ppm TRO throughout the system being protected. Dechlorination is provided for the fossil unit when shock treatment is employed.

Chemical treatment (chlorination) was recommended as a principal treatment option of zebra mussel control in all the TVA hydro plant raw water systems studied with the exception of the hydraulic power generation system. TRO for the hydro unit will be maintained below the allowable limits so dechlorination is not required. The physical/mechanical methods, such as scraping, hydroblasting or others, were recommended for use in removing the established mussels in the area which are not protected by the recommended chemical treatment program. Although protection is required, the hydraulic power generation system and its components can withstand some degree of fouling, thus a well timed reactive control strategy may be the best approach. In addition to chlorination, in-line automatic backwash filters were also recommended to control zebra mussel for fire protection systems at Wilson, Pickwick and Wheeler Hydro Plants.

Capital Cost of the facility treatment options ranged from $650,000 - $2.5 million.

Hydropower Development in China

Zhang Boting*

Hydropower Potential

China is abundant in hydropower potential. According to a national survey conducted in 1977-1980, the theoretical hydropower potential in China is about 676GW, the exploitable hydropower potential in China is about 378GW, capable of turning out 1923 TWh of electricity per year. Up to 1993, only about 10% of the hydropower resources is exploited. The hydro potential is mainly concentrated in southwestern China. The upper reachs of the Yangtze River, the Yalutzampo River and the Lancang River in southwest China, possess a great amount of hydropower potential, because they flow down along the edge of the Tibetan Plateau with a big discharge. The Yangtze River is the most powerful river system in China, which possess an exploitable hydro capacity of 197 GW, and will be able to generate 1027TWh of electricity per year. Only the Xilodu and the Three Gorges, the two hydropower projects in planning on the Yangtze River, would have an installed capacity of 15000 MW and 17000 MW respectively. These two hydropower stations will be initiated before 2000. On the other hand, the Hai river and Huai river in north and east China are important rivers for agricultural use, but they have little power potential.

Historical Development

* Engineer, Chinese Society for Hydro-electric Engineering
P.O.Box 2905, Beijing 100761, China

The first hydropower station in China was built in 1910 - 1912, in Yunnan Province. The installation included two units of 240 KVA each. the Francis turbines and the generators were supplied by Voith and Siemens Schuckert of Germany. The project had been expanded several times after it was built. Until 1951, when two 3000 KW units were installed to replace all the old units.

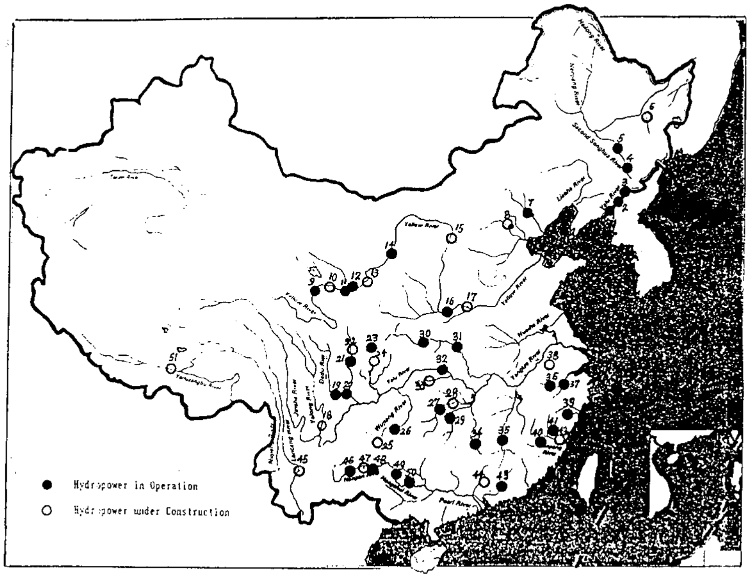
Very little progress was made in hydropower development in the early half of this century. A rapid development of hydropower construction in China began after the founding of the P.R.of China in 1949. The increasing growth records are showed in Table (1)

TABLE(1) PROGRESS IN ELECTRIC POWER DEVELOPMENT IN CHINA

YEAR	CAPACITY TOTAL(Mw)	CAPACITY HYDRO(Mw)	GENERATION TOTAL(GWh)	GENERATION HYDRO(GWh)
1949	1849	163	4310	710
1955	2997	498	12277	2360
1960	11918	1941	59424	7421
1965	15076	3820	67604	10414
1970	23770	6235	115862	20458
1975	43406	13428	195840	47630
1980	65869	20318	300627	58211
1985	87050	26420	410700	92400
1990	137890	36050	621300	126400
1991	151470	37880	677500	124800
1992	170000	40000	746000	132000

In 1992, the national government invested about 10 billion RMB Yuan to the construction of large and medium size (over 25MW) hydropower projects. In this year, the total new hydro capacity commissioned exceeded 2,100MW, not including the 570 MW newly commissioned capacity of small hydropower stations. Large generating units in the Yantan, the Baishan, the Wanan, and in many other major projects were put into

Fig. (1) LOCATION MAP, MAJOR HYDROPOWER PROJECTS IN CHINA



TABLE(2) MAJOR HYDROPOWER PROJECTS IN OPERATOON OR UNDER CONSTRUCTION IN CHINA (1992)

(Legend for Fig.1)

No.	Project Name	Province Located	Capacity (MW)	No	Project Name	Province Located	Capacity (MW)
● 1	Shuifeng	Liaoning	630	● 2	Weiyuan	Jilin	900
● 3	Yunfeng	Jilin	400	● 4	Baishan	Jilin	1500
● 5	Fengman	Jilin	724	○ 6	Lianhua	Heilongjinag	550
● 7	PanjiaKou PS	Hebei	360	○ 8	Ming Tomb PS	Beijing	800
● 9	Longyangxia	Qinghai	1280	○10	Lijiaxia	Qinghai	1600
●11	Liujiaxia	Gansu	1225	●12	Yangoxia	Gansu	392
○13	Daxia	Gansu	300	●14	Qingtongxia	Ningxia	272
○15	Wanjiazai	Shanxi	1080	●16	Sanmenxia	Henan	250
○17	Xiaolangdi	Henan	1560	○18	Ertan	Sichuan	3300
●19	Gongzui	Sichuan	700	●20	Tongjiezi	Sichuan	600
●21	Yuzixi	Sichuan	320	○22	Taipingyi	Sichuan	260
●23	BiKou	Gansu	300	○24	Baozhusi	Sichuan	700
○25	Dongfeng	Guizhou	510	●26	Wujiangdu	Guizhou	630
●27	Fengtan	Hunan	400	○28	Wuqiangxi	Hunan	1200
●29	Zhexi	Hunan	447	●30	AnKang	Shaanxi	800
●31	DanjiangKou	Hubei	900	●32	Gezhouba	Hubei	2715
○33	Geheyan	Hubei	120	●34	Dongjiang	Hunan	500
●35	Wanan	Jiangxi	400	●36	Xinanjiang	Zhejiang	663
●37	Fuchunjiang	Zhejiang	297	○38	Tianhuangping	Zhejiang	1800
●39	Jinshuitan	Zhejiang	300	●40	SaxiKou	Fujian	300
●41	Gutianxi	Fujian	259	○42	ShuiKou	Fujian	1400
●43	Xinfengjiang	Guangdong	293	○44	Guangzhou PS	Guangdong	1200
○45	Manwan	Yunnan	1250	●46	Lubuge	Yunnan	600
○47	Tianshengqiao1	Guizhou	1200	●48	Tianshengqiao2	Guizhou	1320
●49	Yantan	Guangxi	1210	●50	Dahua	Guangxi	400
○51	YangzhuyongPS	Tibet	122				

operation. In 1993, the government investment will be increased by 33 %. Several large projects, such as the 4,200 MW Longtan project, the 1, 260 MW Dachaoshan project, and the 1,800 MW Tianhuangping Pumped Storage project will be initiated. Together with other large and medium size hydropower projects, the capacity of the newly started projects will be over 9,000 MW. The Shuikuo, the Geheyan and the Manwan projects will be put into operation. The newly ommissioned capacity will reach 3,000 MW in 1993. It is predicted that the newly commissioned hydro capacity will be 3,000 MW - 5,000 MW every year after 1993.

In Fig. (1) and table(2), the major hydropower stations, in operation or under construction in China are shown. By the end of 1992, the national installed capacity of hydropower in China has exceeded 40,000 MW, and the annual generation exceeded 130 TWh. The capacity under construction surpasses 40, 000 MW, not including the small (less than 50MW) hydropower projects. Some important hydropower projects and features of China waterpower are describled as follows.

Major Projects

Major hydropower projects in operation or under construction in China are briefly described in the following.

1. The 3,300 MW Ertan project is the largest hydropower station presently under construction on the Yalong river in Sichuan province. The project comprises of a concrete double curvature arch dam, 245 m high, and an underground powerhouse accommodating six 550 MW generating units. The civil engineering works have been contracted to European contractors. Bid invitation has been issued for the supply of generating units in December 1992. The project is scheduled to be commissioned in 1997.

2. The 2715 MW Gezhouba project is the largest hydropower station in operation in China. It is the first project built on the main-stream of the Yangtze river. The project comprises of a barrage dam designed to pass a flood discharge of 83,900 m^3/s, two powerhouses housing two units of 170 MW and 19 units of 125 MW, three shiplocks and other appurtanent works. The project was initiated in 1970, with its first unit commissioned in 1981, and completed in 1986.

3. The 2,000 MW Lijiaxia project has been under construction on the upper Yellow river since 1987. The project has a concrete arch dam 165 m high, and a surface powerhouse accommodating five 400 MW generating units. The generating units are arranged in two rows. Three units in the upstream row and two units in the downstream row. It is a noval design used for the first time in China.

4. The 1,500 MW Baishan project is on the Songhua river in northeast China. The project has two powerhouses, an underground powerhouse in the right bank housing three 300 MW units and a surface powerhouse on the left bank, housing two 300 MW units. The project was completed in 1992.

5. The 1,500 MW Manwan project is constructed on the Lancang river (the upper Mekong) in Yunnan province. Owing to the insufficient space in the narrow river valley, the spillway flip bucket of this project has been designed to run over the top of the powerhouse which contains five 250 MW generating units. First unit of this project is scheduled to be commissioned in June 1993. One more unit will be housed in an underground powerhouse in the right bank.

6. The 1560 MW Xiaolangdi project will be constructed on the Yellow river in Henan province. The project will have a 154 m high rockfilled dam. Fifteen tunnels will be constructed for flood passing, power generation and silt flushing purposes. Construction preparations on this project have been started since 1992.

7. The 1,400 MW Shuikou project has been under construction since 1987 on the Minjing river in Fujian province. The project consists of a concrete gravity dam 101 m high and 786 m in crest length, a surface powerhouse accommodating seven 200 MW generating units, and a shiplock. First unit of this project is scheduled to be commissioned in May 1993.

8. The 1320 MW Tianshengqiao 2 project is situated on the border of Guangxi and Guizhou, on the Hongshui river. The project consists of an intake dam, 58.7 m high, three power tunnels each 9,776 m long, and a surface powerhouse accommodating six 220 MW units. Its first unit was commissioned in December 1992. In the first stage, four 220 MW units will be installed in this project.

9. The 1280 MW Longyangxia project is presently the largest hydropower station in operation on the Yellow river in Qinghai province. The project consists of a 178 m high concrete gravity arch dam, and a surface powerhouse accommodating four 320 MW generating units. The reservoir formed behind the dam has a gross capacity of 24. 7 billion m3. This large reservoir provides huge benefits for the downstream projects in a significant increase in firm power production, a drastic reduction in flood flow and a remarkable increase in irrigation water supply. The project was initiated in 1976, with its first unit commissioned in 1987, and all the four units in operation in 1989.

10. The 1,225 MW Liujiaxia project is situated on the upper Yellow river about 90 km from Lanzhou. The project consists of an concrete gravity dam 147 m in maximum height and a powerhouse complex wnich incorporates a surface powerhouse at the dam toe and an underground powerhouse in the right bank. The surface powerhouse accommodates three generating units, while the underground powerhouse accommodates two generating units. The two powerhouses were connected into one machine hall. The project was initiated in 1958 and was commissioned in 1969.

11. The 1,210 MW Yantan project is situated on the Hongshui river in Guangxi Autonomous region. The project consists of a concrete gravity dam 106 m high, a powerhouse and a shiplock. Construction on this project was started in 1985, and its first unit was commissioned in September 1992. RCC work was used in this project in the 52.3 m high cofferdam as well as the 106 m high spillway dam.

12. The 1,200 MW Geheyan project is constructed on the Qinjiang river in Hubei province. The project consists of a concrete gravity arch dam 151 m in maximum height, a surface powerhouse accommodating four 300 MW units and a shiplift. Construction on this project was started in 1986. The first unit of the Geheyan project is scheduled to be commissioned in July 1993.

Pumped Storage

Pumped-storage is a new addition to China's electric power grids, and its role is increasingly important. Since the economic reform proclaimed by the

government in 1978, electric power demand has been increasing at an average anuual rate of 7 %. With the improvements in people's living conditions, electricity consumed by residential consumers has been increasing at an even faster rate (15 % annually) . Electric lighting, for example, which is the most basic use of electricity in China, has been increasing at an annual rate of 25 % in some large cities in recent years. This resulted in a rapid increase in peak power demand. Statistics show that the electricity consumed for residential uses increased by 10 % between 1986 and 1990. The share of residential load represented 5.5 % of the total electricity consumption in 1986, and increased to 7.5% in 1990. The total electricity consumed in residential use was 50 TWh in 1990, and is predicted to increase to 200 TWh by the year 2000. This figure is still at a very low level compared with international standards, for a country with 1.1 billion population. In addition, the industry and the economic growth is much faster in the coastal regions where no more large hydropower stations can be built. To meet the rapid increase of peak power, pumped storage staions were initiated in recent years. At present there are 5 pumped-storage projects in operation or under construction in China. They are;

<u>The 420 MW Panjiakuo pumped storage station</u>, with 1 conventional 150 MW unit and 3 reversible pumped storage 90MW units, completed in 1992. Its average annual generation is 589GWh, with 238 GWh from conventional unit and the rest from pumped storage units.

<u>The 1, 200 MW Guangzhou Pumped-Storage Station</u> is located in Guangdong Province, 90 km northeast of Guangzhou city. It will be run in paralled with the 1,800 MW nuclear power station at Daya Buy, under construction near by. The average head used is 530 m. Four 300 MW pumping generating units are being installed, and three of them are scheduled to be commissioned in 1993. Besides, a feasibility report has been completed and approved to build a second stage of the Guangzhou Pumped-Storage project, which would add a further 1200 MW capacity.

<u>The 800 MW Ming Tombs pumped-storage project</u> is at the Ming Tombs reservoir, 40 km north of Beijing city. The pump turbines will be of the reversible Francis type, to be operated at a rated head of 430 m for generation and 446 m for pumping. The motor-generators will have a rated output of 220 MW as a generator and 225MW as

a pump. According to construction schedule, the station will be commissioned by 1995.

The 1, 800 MW Tianhuangping pumped-storage project is located in northern Zhejing in the heart of the Yangtze Delta. The site is 170 km from Shanghai, 57 km from Nanking, and 40 km from Hangzhou. Many large load centres are concentrated in this area. In this region, the East China Power grid provides power supply to a population of 130 million. The power grid is expanding at an annual rate of 10% . At present, hydropower takes only 10 % of the grid capacity. Peaking capacity is urgently needed. The average head between the upper and lower reservoirs is 566 m, six 300 MW reversible units will be installed for generating and pumping.

The Yangzhuyong Lake PS. project is located in Tibet, 83 km fromLhasa City. 4 pumping and 1 conventional generators, each 22.5MW, will be installed, working head 849m, with annual generation 80-110 MWh. The station will be in operation in 1994.

Small and Mini Hydropower

The rural country of China is inhabited by 870 million people. About 1/3 of the 2,200 counties in China rely on small hydro for their electricity supply. By the end of 1992, about 55,000 small hydro stations are operating in China. All of them were built by the rural people themselves at low construction cost, by the use of local material, domestic made equipment and rural labour. There are100 factories in China specialized to produce small hydro machinery, with a total production capacity is about 1,000 MW a year. 30 years ago, the government used to supply the generating equipment to the small hydropower station free of charge. Later on, a government grant equivalent to 1/3—1/6 of the project cost was provided as a policy guiding investment. Recently, government grants are replaced by low interest loans which can be obtained from the agricultural bank. In order to set up leading examples so as to stimulate rural electrification in the whole country, a rural electrification programme has been launched by the Chinese government since the early 1980s. The first programme covered 109 countres, and was implemented in 1984 - 1989. The main achievements of the programme include;

- A total installation of 2446 MW of small hydro was reached in the 109

counties by June 1990.

- Their total industrial and agricultural product value increased by 128 % in the five years.
- The average annual per capita income of the inhabitants increased by 304 %, reaching 620 RMB Yuan or US$ 115/capita/year.
- The central government invested 500 million RMB Yuan to the programme.

the first programme fulfiled in schedule in 1990, benefiting 109 counties with 30.2 million inhabitants. The countres selected are mostly in the poor, remote and mountaineous areas. the region inhabited by national minorities enjoyed a priority to be included. The introduction of electricity to the rural community has made a great change in the social and economic life of the peasants. Many undertakings, such as pumping irrigation, the processing of the agricultural products and sideline production in the rural families, were improved much more. Daily activities could be extended for several hours in the evening, because of the improvement of the lighting condition. Besides, films and other entertainment, especially TV programme give the inhabitants much more enjoyment in their cultural life. Now, the second programme to be implemented in 1991-1995 is underway. 200 counties with about 70 million people will benefitted. By the end of 1992, the total installed capacity of the small hydro stations in operation in China reached14,300MW, and producing 40 TWh electricity annualy.

With the development of rural economy in China, mini hydro sets designed and manufactured for family use have been booming in the region where no electricity is available, since the middle of 1980s. Mini hydro sets in 650W to 2kW range are being producted in 50 factories in seven provinces in southern China. Two types of turbines are available; the propellor type are designed to be used for a head of 2 m to 4 m, the turgo type turbines are designed for a head of up to 30 m. the turbines are usually connected to a condensor type asynchronised generator, which produces single phase AC currents at 230 Volts. The machinery is usually made in packaged form, with the turbine and the generator assembled together weighing no more than 30 - 50 kg, and can be easily installed following simple instructions with the minimum amount of civil work. The station can be constructed by the family members in 7-10 days. The cost of a mini hydro set is about US$ 300-500/Kw. The mini hydro set is

usually owned, installed and operated by one family or a few families, and supplies electricity to one or a few families or a small village. By the end of the 1991, there were 28, 000 mini hydro sets in use to a total capacity of 33, 500 kw, distributed in 10 provinces in China.

Conclusion

According to China's economic development programme, the total electric power capacity will reach 240,000 MW by the year 2000, among which hydropower would be about 80,000MW . In order to realize this target, the hydropower will be developed more rapidly in China in the coming years. By the end of 1993, the capacity of hydropower under construction will reach 50,000 MW. Advanced techniques will be used in large scale projects, for example, the Roller-Compacted Concret dam in Longtan (H=216 m), the Concrete-Faced Rockfill dam (H=180 m) at Tianshenqiao 1, and the Thin Double-Arch Concrete dam at Ertan project(H=245 m). In addition, the largest hydropower project,The Three Gorges, will be put into construction in the near future. Generally Speaking, Chinas' hydropower will be developed in a higher tempo in the following decade.

Monthly Waterpower Dependability

Colby V. Ardis[1] and Aaron A. Jennings[2]

Abstract

The conventional technique of determining a frequency distribution of average daily discharge to evaluate run-of-river waterpower dependability at a potential power site may overestimate dependability. The use of annual statistics desensitizes the analysis to monthly variations that occur during likely sub-annual operating periods. For example, the statistical nature of June flows may be significantly different than those of December. Hydropower projects that operate throughout the year experience all of these flows. Further, projects operated for only select portions of the year should be evaluated with the appropriate sub-annual statistics. Analyses based on annual statistics may overestimate the potential value of partial-year operation because the data for water-abundant periods skews results toward large flows that may not be representative of the actual period of operation.

Introduction

The probabilistic stage and flow properties of water resources are essential considerations to the analysis of potential run-of-river power projects. These are important considerations to most water resource projects, but run-of-river projects "live" or "die" on a day-to-day basis. If the volumetric flux (Q) or differential head (ΔH) are inadequate, run-of-river projects cannot satisfy the demand for which they were designed. Less dependable power may be used on an "as-available" basis, but this will be less valuable. The viability of many run-of-river projects may hinge on how accurately the "firm" power properties of the site are analyzed.

----

1. Dean, College of Engineering, Southern Illinois University at Edwardsville, Edwardsville, IL, 62026-1804.
2. Professor, Department of Civil Engineering The University of Toledo, Toledo, OH 43606.

A great deal of theoretical and applied work is available to assist engineers in estimating the stage and flow properties of water resources. The United States Geological Survey has developed a huge body of data from gauging stations on the major water courses of the United States. Mature statistical methods are available for translating this data into stage and flow predictions. Most states have also developed procedures that allow engineers to estimate flows and/or stages at ungauged sites. However, nearly all of this focuses on the analysis of statistics that associate values with time periods of at least a year. In this paper we will explore the analysis and implications of sub-annual project periods.

Most of the probabilistic analyses done for water resource projects are based on annual time periods because most water resource projects experience the long-term hydrologic extremes of the resource. Flood control dikes, for example, stand ready year after year for their opportunity to protect communities from hydrologic extremes. It makes sense to design such dikes using the distribution of the annual maximum flows. However, this is not the case for all projects. Consider a run-of-river project designed to supply hydropower at a remote summer recreational site. If the stream's statistical hydrologic response is dominated by fall or spring flows, the power capacity of this site should not be evaluated using annual statistics. These would overestimate the total power of the actual sub-annual project life, and would overestimate the "firm" power rating based on the minimum expected flow because the large monthly values would make the flow minimums seem less likely than they would actually be. There are several types of water use projects that do not experience the full annual water cycle. Analysis based on sub-annual statistics may yield more realistic and cost effective designs or operational strategies for these.

It is not the authors' intention to suggest that most water resource projects should be analyzed using sub-annual statistics. Annual analysis will remain an essential component of water resources engineering. However, we do hope to illustrate that the statistical properties at many sites can be considerably different when viewed over sub-annual periods. This variability creates the potential for engineering blunders such as over design or non-optimal operation, and creates opportunities for creative use of what are becoming increasingly scarce water resources.

Flow Analysis Based on Annual Statistics

Several federal agencies have attempted to promote a consistent approach to flow frequency analysis. Although many people feel that no single technique should be used exclusively because one cannot know a-priori which theoretical probability function best describes randomness

at a particular location, there are advantages to standardization even if the standard is fallible. The technique described in Water Resources Council Bulletin No.15 (WRC, 1967) was originally advocated for use in all federal planning. The procedure was revised in Bulletin No. 17 (WRC, 1976) and modified in Bulletins 17A and 17B (WRC, 1977, 1981). The Water Resources Council ceased to exist in 1982, but its Hydrology Subcommittee continued as an interagency advisory committee, and continued to advocate use of Bulletin 17B (Tasker and Stedinger, 1986).

The method of Bulletin 17B is based on the Pearson Type III (LP3) distribution applied to $\log_{10}$-transformed data. The LP3 distribution subsumes the classical Gamma probability density function (PDF) and has been applied extensively to hydrologic events. Although this selection has been criticized, and alternatives such as the Gumbel, Pareto and Weibull distributions have been advocated, most recent discussions have focussed on the difficulties of site-specific calibration of the LP3 distribution. These arise because the LP3 requires accurate values for the first three moments of the random variable. The typical application to annual hydrologic data limits the number of points available. Although some locations may have hydrologic records of 100 or more years, the systematic impacts of man often preclude use of older information. Locations with more that 30 or 40 years of dependable data are rare. Therefore, although the mean and variance of the flow or stage can often be estimated accurately, it is more difficult to estimate values for the third moment (skewness). This is troublesome because the LP3 distribution is sensitive to skewness, particularly in the neighborhood of extreme values.

Bulletin 17B recommends the method of moments for establishing the essential coefficients of the LP3 distribution.

$$\overline{X} = \frac{1}{N}\sum_{i=1}^{N} X_i \; ; \quad S^2 = \frac{1}{N-1}\sum_{i=1}^{N}(X_i - \overline{X})^2 \; ; \quad G_s = \frac{\sum_{i=1}^{N}(X_i - \overline{X})^3}{(N-1)(N-2)S^3}$$

Here X_i are the N transformed data, X is the sample mean, S is the variance and G_s is the station skewness. Because the estimate of G_s is subject to large errors, use of "regional" skewness weighting $G_W = WG_S + (1\text{-}W)G_r$ is recommended. Here W is the weighting factor and G_r is the "regional" skewness value. Generally the weight is inversely proportional to the mean square errors ($W = (MSE_r)/(MSE_r + MSE_S)$) of the skewness estimates. Regional skewness and MSE_r values may be computed directly but most often they are interpolated from a generalized skew contour map of the United States (see Fig. 1) prepared for this purpose (WRC, 1976). When skewness is interpolated from this map, the value of MSE_r is 0.302.

Several authors have noted that the contours of the Generalized Skew Contour Map do not fit particularly well in some states. For example Fig. 2 presents an expanded view of the map for the state of Ohio. Note that, although the contours seem to fit reasonably well in Northwestern Ohio, they do not capture skewness properties in the central portion of the state.

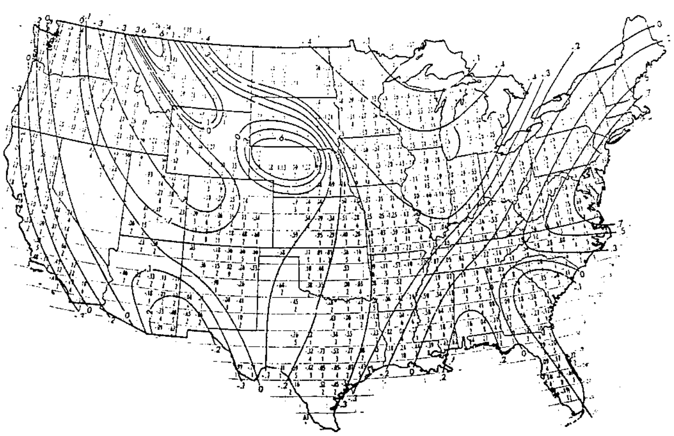


Figure 1 - National Skewness Contours (WRB, 1976)

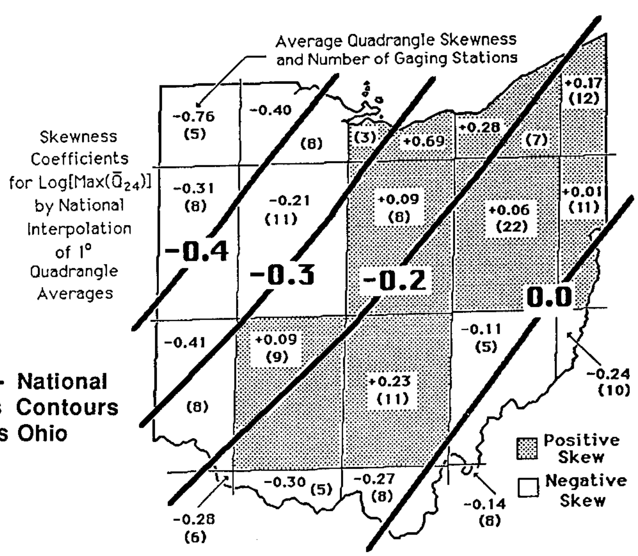


Figure 2 - National Skewness Contours Across Ohio

Flow Analysis Based on Sub-Annual Statistics

Using the Bulletin 17B procedure for associating probabilities with flows, we will illustrate the potential ramifications for projects that function for sub-annual periods. For the sake of compatibility with USGS data , we will consider sub-annual periods corresponding to one or more consecutive but complete calender months. Also, for the purpose of illustration, we will only consider data from the state of Ohio.

Given a location in Ohio, one can approximate the annual regional skewness value from Fig. 1 or 2. Is it reasonable to expect that this value is appropriate for a sub-annual period of the year ? The answer appears to be no. Figure 3 presents the results of a month by month analysis of station skewness for 16 gauging stations in Ohio. The 16 stations were selected for completeness of their flow record and to sample small, medium, and large water courses across the hydrologic regimes of the state. As one might expect, there is a wide range of skewness values for these stations. However, the most notable feature of Fig. 3 is a distinct annual pattern of skewness that varies from negative during the winter months, to positive during the summer. This pattern exists in the average skewness values, and in the upper and lower bounds of the individual station extremes. This annual pattern appears to be a real feature of the hydrology of Ohio. There are also substantial differences in the monthly means and variances of the data for these stations, but the station skewness data best emphasize the fact that the probability distribution itself also varies during the annual hydrologic cycle.

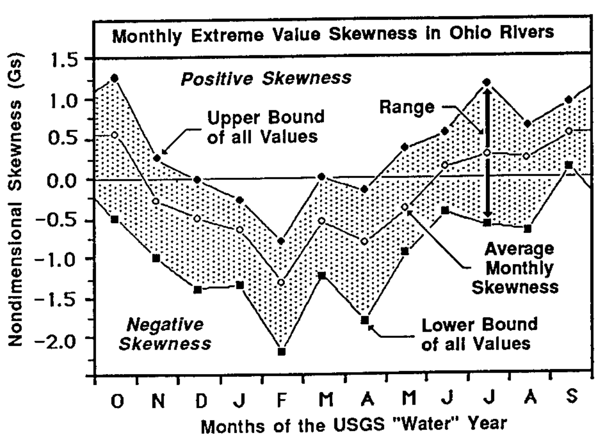


Figure 3 - Annual Variation in Log(Discharge) Skewness of the Monthly Extreme Flows in 16 Ohio Rivers

To illustrate the potential significance of this annual progression, Table 1 presents the results of defining sub-annual periods of one month on the estimate of the 2, 5, 10, 25, 50, and 100 year recurrence interval events for the Clear Creek of southeastern Ohio. Clear Creek is in the Hocking River Basin with a drainage area of approximately 89 mi^2. Flow has been gauged at this location since October, 1939. The creek has an annual average discharge of 90 cfs, and a historical maximum flow of 16,000 cfs.

Table 1-Sub-Annual One-Month Estimates of Maximum Flows Compared to Annual Estimate of Maximum Flow for Clear Creek at Rockbridge, OH.

Month	M-Year Recurrence Interval of Extreme Value Flow Estimate					
	2	5	10	25	50	100
October	49.	93.	159.	304.	487.	768.
November	100.	240.	387.	654.	925.	1271.
December	239.	619.	984.	1576.	2110.	2719.
January	428.	926.	1327.	1885.	2324.	2776.
February	577.	1145.	1508.	1912.	2168.	2387.
March	582.	1318.	1965.	2947.	3786.	4709.
April	521.	1052.	1470.	2053.	2515.	2995.
May	298.	711.	1162.	2023.	2942.	4166.
June	230.	508.	789.	1287.	1786.	2418.
July	134.	395.	765.	1669.	2882.	4850.
August	107.	325.	619.	1289.	2127.	3399.
September	54.	148.	276.	582.	985.	1631.
Annual	2550.	4230.	5860.	8720.	11590.	15280.

The values of Table 1 are based on a modified definition of recurrence interval. The M-year recurrence intervals of Table 1 indicate the flow that, on average would be exceeded once in M years by a daily maximum flow within the sub-annual time period. For comparison, Table 1 also includes the recurrence interval flow estimates based on the conventional annual period. All Table 1 values were calculated using the LP3 distribution.

The numbers of Table 1 indicate dramatic differences between extreme values based on sub-annual and annual periods. For example, the 2-year flow (i.e. flow exceeded 50% of the time) in September is only about 2% of the value based on annual statistics. In fact, all of the values of Table 1 are significantly lower than the annual estimates. This may seem counter-intuitive since all of the extreme values used to make the annual predictions are also represented in the monthly predictions. However, when the monthly extremes are determined, the annual extreme is used for the month in which it occurs. If the annual extreme occurred in the same month each year, then the statistics for that month would be identical to the annual values. Although the annual extreme often occurs in March for Clear Creek,

it can occur in any month. Therefore, the extreme value set for any month may contain some annual extremes, but these are "diluted" with less severe monthly extreme values. These monthly extremes make the annual values seem much less common which is correct from the monthly perspective.

Consider the possible run-of-river power ramifications of these calculations for a project designed to yield power during the summer months of June, July and August. A total power estimate based on the annual average maximum of 2,550 cfs would greatly overestimate the true hydropower potential since, during the sub-annual project period, no monthly average maximum exceeds 230 cfs. For this particular problem, an analysis based on all average daily flows during the sub-annual operation should be done to determine the site's "true" firm power rating.

Summary and Conclusions

Conventional statistical analysis of hydrologic data is generally based on annual statistics. For many applications this is appropriate, but the results may be misleading if the project is designed for sub-annual operation (Ardis,1986). Sub-annual operational periods do not experience the full hydrologic cycle. They may not experience the spring "floods" or the summer "droughts" typical of many drainage basins. Therefore, the probabilistic analysis of the flows the projects will actually experience must be adjusted to consider the correct subset of the annual response. In the example presented, we have illustrated that when monthly subsets are considered, there can be surprisingly large deviations in the probabilistic response. We have demonstrated this using large flow data, but similar results (i.e. big differences) would be associated with the other end of the distribution. This investigation is underway. From our work to date, we conclude that the statistical response of sub-annual subsets of the "typical" hydrologic year should be considered for all projects that do not operate over the whole hydrologic cycle.

References

1. Ardis, C.V. (1986), "Advantages of Monthly Flood Risk Analysis Models to the Surface Mining Industry", National Symposium on Surface Mining Hydrology, Sedimentation, and Reclamation, Univ. of Kentucky, 235-238.
2. Tasker G.D. and Stedinger, J.R (1986), "Regional Skew with Weighted LS Regression", **J. Water Res. Plan. and Mgmt.**, 112(2), 225-237.
3. Water Resources Council (1967), A Univorm Technique for Determining Flood Flow Frequencies, Bulletin No. 15.
4. Water Resources Council (1976), Guidelines for Determining Flood Flow Frequency, Bulletin No. 17.
5. Water Resources Council (1977), Guidelines for Determining Flood Flow Frequency, Bulletin No. 17A.
6. Water Resources Council (1981), Guidelines for Determining Flood Flow Frequency, Bulletin No. 17B.

HYDROPOWER IN BRAZIL: PAST VIEW AND FUTURE DIRECTIONS

Paulo Sergio Franco Barbosa*

ABSTRACT

In this paper the main steps of Brazilian development in the area of hydroelectricity are presented. The historic evolution is illustrated and discussed by means pertinent data about hydropower and consumptiom of electricity in Brazil. A short description about the modern scenario and main future challenges are also included.

INTRODUCTION

Brazil has the third largest hydroelectric potential in the world. From the currently estimated at 213 GW for installed capacity, only 26.2% of this total has been developed. Despite this great potential, one of the main difficulties posed to the Brazilian planners comes from the geographic distribution of natural resources for hydrogeneration, not suitable to distribution of the demand. The future projects of lower cost/KW will tend to be located far from the main load centers and long-distance transmission of huge power blocks will have to be taken into account.

The investiments costs for the big projects are high, and there are few opportunities of getting financial support abroad. Furthermore, the feasible taxes for electricity consumptiom has the lower purchasing power of Brazilian people as a serious limit. In this scenario, the expansion of hydroelectric generation is also restricted by growing awareness of Brazilian society concerning to environmental issues. After a presentation of a brief retrospect about the historical evolution of hydropower in Brazil, the new trends and alternatives are discussed.

*Ph.D., Assistant Professor, Department of Civil Engineering, State University of Campinas-Unicamp, P.O. Box 6021, CEP 13081-970, Campinas-SP, Brazil

THE EVOLUTION OF HYDROPOWER IN BRAZIL

The great drougth in the Northeast region during the period 1877-1880 marks the beginning of dam construction in Brazil. Almost at the same time the first dam for a hydroelectric development (Ribeirao do Inferno) was built in the state of Minas Gerais in 1883 by a private company. The next hydropower plant (Marmelo) was built in 1889, with the electricity being used for public services. According to Mielnik and Neves (1988), the evolution of installed capacity around the end of XIX century occured in the following way: 52 KW in 1883, 12,085 KW in 1900 (46 % from hydropower) and 159,890 KW in 1910 (86% from hydropower).

At Table 1, excerpted from Holtz (1985), it can be noted the rate of growth of total built hydroelectric plants in the country. The construction of hydroelectric plants was accelerated especially after the 1950s. After the Second World War, Brazil experienced an initially modest, but quickly progressing development of industrial activities. During this period of economic growth, the yearly increase in demand for electrical energy of up to 12% was well above the world average. New hydro plants with ever larger dams began to be constructed in regions more and more distant from consumptiom centers when the available hydro potential in the sorroundings of the latter became exhausted. Because of the increasing importance of the energy sector for the nation's economy, the Ministry of Mines and Energy was created in 1960. From then on it handled the licensing of new hydropower developments and delegated to Eletrobras, the government owned holding company of federal interests in the field of electric energy production, the coordination of power planning nationwide.

At present, there are in the country around 300 hydraulic plants in operation or under construction, from which 11.3% have power capacity above 400 MW, 12.3% between 40-400 MW and, 76.4% lower than 40 MW. Fewer plants began to be operated in 1976-85 and in 1986-90 than in previous ten-years periods. The main reason for this is that those being built have a much larger power capacity. Taking Itaipu (12,600 MW) and Tucurui (3,980 MW) as examples, the magnitude of these two enterprises can be better appreciated when one compares their respectives capacities, not only with the total installed in the country (55,897 MW), but also with other hydroelectric plants in the world.

Gradually the construction of these hydro plants has introduced in Brazil the large-scale plant engineering concept, and the elaborate technology associated with the existing industrial structure. This at first led to a reliance on imported equipment and engineering services, construction, and assembly, and a number of heavy manufacturers were established. After 1975 the evolution of Brazilian engineering has allowed a significant local participation in the main stages of a hydroelectric enterprise: planning and design; construction and; production of electrical and mechanical equipments. Now the Brazilian civil construction in-

Decade	Number of Plants		Installed Capacity (MW)	
	In the Period	Cumulative	In the Period	Cumulative
1936-1945	16	99	211	955
1946-1955	44	143	965	1,920
1956-1965	56	199	4,685	6,605
1966-1975	53	252	12,183	18,788
1976-1985	31	283	19,917	38,705
1986-1990	7	290	16,677	55,382

Table 1: Number and Installed Capacity of Hydroelectric Plants in Brazil

dustry has reached all-time high production volumes for major Brazilians dams, with 1.235 millions of m^3 /month of rock excavation, 1.705 millions of m^3/month of earth excavation, 337,000 m^3/month of concrete laying, 896,700 m^3/month of rock filling, and 2.497 millions of m^3/month of earth filling. Furthermore, the local heavy equipment industry is capable of producing, among other things, up to 97 GW/year of hydroelectric turbines, up to 8,900 MVA/year of hydrogenerators, and up to 35,000 MVA/year of transformers and reactors.

THE MODERN SCENARIO AND FUTURE CHALLENGES

Some Characteristics of Brazilian Hydroelectric System

In order to get a better understanding about the nature and the growth of consumptiom of electricity in Brazil, some pertinent data are shown in Tables 2 and 3, which were excerpted from Santos (1992).

Year	Residential	Commercial	Industrial	Others	Total
1987	38,378.9	20,455.0	95,813.8	25,787.6	180,435.3
1991	51,078.2	24,912.6	102,920.1	30,687.8	209,598.7
1992	54,309.0	26,218.0	107,484.0	32,377.0	220,388.0

Table 2: Consumptiom of Electricity in Brazil (GWh)

In Table 3, it can be noted the greater rates of growth occured in decade of 70. The governnamental policies for the enlargement of hydropower have the petroleum crisis as one of the main sources. Furthermore, the financial resources obtained from developed countries were often applied in construction of large hydropower plants. Therefore an intensive and progressive feedback between a

large installed capacity and actual needs of energy have increased the consuptiom of electricity at high annual rates up to 14% registered in 1976. However, this situation have suffered significant changes since 1980. The decline of national economy was the main reason for this reduction.

Years of 1970 Decade	Rate of Growth (%)	Years of 1980 Decade	Rate of Growth (%)
1971	13.0	1981	2.0
1972	11.0	1982	7.0
1973	13.0	1983	6.0
1974	11.0	1984	10.0
1975	10.0	1985	8.0
1976	14.0	1986	4.0
1977	12.0	1987	0.5
1978	12.0	1988	8.0
1979	12.0	1989	5.0
1980	10.0	1990	1.9

Table 3: Evolution of Consumptiom of Electricity in Brazil

Any analysis of alternatives for electricity supply in Brazil must consider the geographic distribution of potentially capacity, which is estimated in the following way: 97.8 GW (45.9%) in North/Central region; 15.5 GW (7.3%) in North East region; 56.2 GW (26.2%) in South East/ Central West region and; 43.5 GW (20.6%) in South region. This distribution has the largest portion still available in the North/Central West and South, especially in the North, where roughly 97.8 GW remain to be developed. In constrast with this concentrated remaining potential, the more intensive industry and greater cities are located in South East and South regions (see table 4), where it was reached a development rate of 46.7 per cent.

Year	North/North East	South/Southeast	Total
1987	32,995.7	147,439.6	180,435.3
1991	44,212.4	165,386.3	209,598.7
1992	47,569.0	172,819.0	220,388.0

Table 4: Consumptiom of Electricity by Regions in Brazil - GWh

Nowdays there have been few investiments for expansion of hydropower plants, as can be noted in Table 5 that shows the recent decline of annual growth of installed capacity . However, in spite of geographic disparities, progress made locally in the technology of long-distance electric energy transmission has made

it economically feasible to set up, in the future, a single interconnected system for all Brazil. Then, it would enable the country antecipate development of the northern potential to the benefit of other regions. Now an interconnected transmission is available for South/South East rgions and also for North/North East regions. The planning of operation for the electrical systems based on these integrated regions have been allowed to get better use and efficiency of installed capacity. Some aspects of this subject will be presented below.

Year	Installed Capacity (MW)	Annual Rate of Growth (%)	Total Length of Transmission (Km)	Annual Rate of Growth (%)
1986	42,960	—	47,755	—
1987	46,450	8.1	49,985	4.7
1988	49,070	5.6	52,146	4.3
1989	53,046	9.2	54,377	4.3
1990	54,744	3.2	55,095	1.3
1991	55,897	2.1	55,588	0.9

Table 5: Growth of Capacity and Length of Transmission System in Brazil

The Operation Planning of Brazilian Electric System

State companies constitue the main sector for generation of electricity in Brazil. The general principles and rules for planning and operation are established by Eletrobras, a federal holding company created in 1962. The growing interconnection among the electric system of the various Brazilian regions requires carefull coordination among companies. This task is made by Eletrobras, through a specific group named GCOI (Coordinator Group of Interconnected Operation).

The operation planning of the Brazilian hydrothermal system poses some particular issues not common in other countries. The entire system is hydrodominant, with hydropower corresponding to 92% of installed capacity. Despite the interconnection along the country, there is no connection with another thermal system abroad. Therefore, the operation planning is significantly relied on seasonality of river flows. The energetic planning has the reliability of supply as the major objective, being adopted a 5% as an usual value for the risk of shortage. Another specific characteristic is the large size of the systems, with a great number of power plants in a same river basin. Taking the hydrosystem of Parana river basin as example. There are 32 hydro plants in this basin, including a large range of hydropower kinds. A third characteristic of Brazilian hydropower system is the great randoness of river flows, since the unique contributing factor is rainfall. Thus, the stochastic nature of river flows must taken into account in operation planning.

In order to have a better use of available information, operation planning procedures can be broadly divided into three main stages: (a) long-term planning (5 years); (b) medium-term planning (1 year) and; (c) short-term planning (1month).

The long-term planning (LTP) comprises the analysis of the behavior of the system for different hydrologic conditions, delays in construction and other scenarios. Aiming to emphasize the stochastic nature of the problem, it is adopted the composite representation of all reservoirs in each region. This aggregation is performed in order to allow the use of stochastic dynamic programming in the hydrothermal optimization (Ferreira et alli., 1986).

The medium-term planning is corcerned with the definition of operation policies for one year ahead. Once an optimal operating police is obtained from LTP, the system is simulated in a disaggregated basis in order to obtain the storage targets of each reservoir. Studies consider different hydrologic conditions and, it should taken more detailed aspects into account such as: contracts for energy and power exchange between utilities; expected fuel consumption; maintenance scheduling, etc.

The short-term planning establishes the generation targets of each plant during the period. Very detailed studies of the system operation are required such as daily flood control rules and simplified simulation of the hourly power flow along the week. Load and river flow forecasts are extensively used.

Besides hydropower, flood control has also a significant role in reservoir operation of Brazilian system. In fact, the reservoirs of hydroelectric system were not designed considering flood control. However, the resulting river-flow regulation has made the downstream population overconfident on the capability of the reservoirs to mitigate floods with moderate return periods. It has become necessary to adjust the operating rules in order to take flood controls constraints into account.

Multipurpose Use of Dams and Environmental Issues

The first Brazilian dams and reservoirs have been planned and brought into service for single purposes. Until the 1950s, the private power industry as well as government agencies assigned to irrigation and flood control protection projects only took into consideration the requirements and necessities of their own activities when designing new dams. From then on, however, the steadily growing demand for electric energy and the booming construction of hydro plants emphasized the necessity for coordinated planning. It was the power industry – more and more under the control of the federal and state governments – that initiated, in the early sixties, large-scale inventory campaigns and development planning of hydraulic resources for power production. However, neither the multipurpose use of reservoirs nor environmental aspects were taken into consideration at that time. It took almost another decade until public concern started to be aroused, when the results of massive pollution of surface waters by municipal

and industrial waste became obvious. Soon the question was raised among those responsible for the design and operation of dams as to how damming itself was going to affect the environment. Thus, in 1972, the Brazilian Committee on Large Dams started dealing systematically with the relationship of dams and the environment, and the problems involved.

The fast economic growth during the sixties and seventies resulted in the need to tap the vast reserves of water resources of the country's interior. Longer distance of power transmission and the additional financial burden associated with the work at remote sites led to a rapid escalation in construction and operating costs. As a result, increasing attention was paid to how these costly projects could be better used by having more than one function. There was a growing awareness that the available water resources were not inexhaustible, and that they would have to be used wisely to satisfy future demands for irrigation, water supply and river navigation besides power generation, without unnecessary disturbance of natural environment.

An additional point of evolution in Brazilian scenario was concerned with the companies that have been exploiting hydropower. All major Brazilian power companies are owned either by the federal or by a state government. Thus, besides the production and sale of electricity, they have been acted as agents for the government in local and regional devlopment programmes linked to the construction of new dams. For this reason, reservoir planning is often extended to cover items beyond the immediate use of the reservoir.

In few years ago the Tucurui hydro plant, located at the Amazonia river basin, was a specific project that required great efforts for evaluation of environmental impacts. As better described by Monosowski (1983), it was the largest man-made lake in Brazil, and the largest one constructed in a tropical rainforest. Tucurui dam created a 2160 km^2 reservoir, flooding the forest and backing up water for 200 Km along Tocantins river. The accumulation of 43 millions of m^3 of water allowed a capacity of 3960 MW in a first stage, available since 1985, for a total capacity of 7260 MW. The environmental impacts of Tucurui project were difficult to assess: the Amazonian ecosystem are very complex, diverse and almost unknown. In addition, the advance of economic frontier, stimulated by major projects, is rapidly causing environmental changes in the whole region.

Despite the controversy evoked by Tucurui project, this experience was very valuable for analysing the possible future projects in the Amazon. According to Holtz (1985), hydroelectric inventory studies conducted in the this region indicate that more than 50% of the Amazon basin potential could be concentrated in a few large plants at a relatively low cost (less than US$ 30/MWh). Placing the energy produced by these plants in the Southeast and Northeast areas by large transmission trunk lines would represent an additional cost of roughly US$ 10/MWh. This means that the final cost would be competitive with most smaller hydroelectric plants to be built in the Southeast, and with coal or nuclear thermoelectric plants. Refinement of such studies, as well as a more acquaintance with thermoelectric plants, and with the question of choos-

ing between large hydroelectric plants in the Amazon or medium-size ones in the Southeast and South, coupled with financial variables, will condition the decision on the future expansion of the Brazilian electric system.

CONCLUSIONS

The historic evolution of hydropower in Brazil, with some few exceptions, has showed good results in terms of the development of technology in the area of hydroelectricity and use of water resources.

After periods of high annual rates of growth, occured mainly in decade of 1970, the future growth rates for hydropower will depend on several factors like: growth of the electric energy market, technology acquisition policies, financing facilities for developing available resources, and also on the introduction of pumped-storage plants which are still not taken into account in the aforementioned potential. Complete development of Brazilian hydro potential will occur within a period of time which will depend on the country's economy growth and on future policy options with regard to other primary energy sources. At present, it is difficult to predict how the Brazilian economy will develop in the context of a world economy susceptible to changes brought about, among others things, by a concern with the efficient use of energy and its ecological aspects.

Preserving the environment, or at least minimizing negative ecological effects, has played an increasingly important role in the selection of differents sources of power supply, and is likely to become a major conditioning factor for the energy model to be adopted in the future. However, hydroelectricity is expected to play a major role for many decades to come in the Brazilian electric supply and, in the author's opinion, the country is prepared to develop its full potential in the technological, managerial and industrial view-point.

ACKNOWLEDGMENT

This work is based on research experience of the author in the area of hydroelectricity, which have been supported by FAPESP - Fundacao de Amparo a Pesquisa do Estado de Sao Paulo.

REFERENCES

Ferreira, C., Barreto, L.A.L., Araripe Neto, T.A., Fortunato, L.A.M. (1986), "Energy Operation Planning of the Brazilian Interconnected System", Study Committee 39, *CIGRE Meeting*, paper n.39-03, Rio de Janeiro.

Holtz, A.C. (1985) "The Future of Hydropower in Brazil", *Water Power and Dam Construction*, January, pp.16-19.

Mielnik, O., Neves, C.C. (1988) "Caracteristicas da Estrutura de Producao de Energia Hidreletrica no Brasil", pp.17-38 in *Impactos de Grandes Projetos Hidreletricos e Nucleares*, ed. Marco Zero, Rio de Janeiro, Brazil.

Monosowski, E. (1983) "The Tucurui Experience", *Water Power and Dam Construction*, July 1983, pp. 11-14.

Santos, M.F.M. (1992) "Operacao dos Sistemas Interligados e Conservacao de Energia no Brasil", *Eletricidade Moderna*, n.216, pp. 67-76.

The Demolition of Mussers Dam

Gerald L. Cross, P.E.[1]

Abstract

At noon on Saturday, November 21, 1992, the demolition of an historic timber frame dam was complete. The following describes the events and procedures that transpired during the demolition of the Mussers Dam.

Introduction

Mussers Dam, a 384-foot long, 31-foot high timber frame and earthfill structure, was successfully breached on October 5, 1992, and the removal of the timber frame portion of the dam was completed on November 21, 1992.

The dam, located about 50 miles north of Harrisburg, in Selingsgrove, Pennsylvania, on Middle Creek, a tributary to the Susquehanna River, was originally constructed in 1906. A flood in 1934, occurring while the dam was being repaired, breached the dam. Mussers Dam was again rebuilt in 1935.

Because of the operation of the hydro power project at Mussers Dam, resulting in frequent variations in the tailwater levels, the Federal Energy Regulatory Commission (FERC) became justifiably concerned that some of the structural members in the lower third of the structure could be in a stage of advanced deterioration. Possible rot, decay, and splitting due to repeated wetting, drying, freezing, and thawing could have greatly reduced the load-bearing capacity of many of these members.

---

[1]Civil Engineer, Federal Energy Regulatory Commission, New York Regional Office, 201 Varick St., Room 664, New York, NY 10014

Since it had been impossible for FERC engineers to properly access the areas of concern to make a visual inspection, and since the Part 12 Inspection by an independent consultant found that the dam had substandard factors of safety, FERC directed the Licensee to perform a thorough structural inspection that would require dewatering of the interior of the dam. In lieu of the structural inspection, the Licensee proposed remedial measures that were deemed unacceptable by FERC. The licensee was then ordered to replace the structure, but since replacement was determined to be economically unfeasible, the licensee proposed to remove the dam and surrender its license.

With the demolition of the dam came the proof that FERC's concerns were warranted. Rotted, decaying, and splitting structural members were found in the lower third of the structure.

When the environmental impacts associated with the removal of the dam were weighed against the safety hazards presented by the unsafe structure, safety was of paramount importance. FERC, therefore, approved the licensee's proposal to breach Mussers Dam on September 3, 1992.

It was understood, and reluctantly accepted by the various environmental agencies prior to the beginning of demolition, that there would be a large amount of silt erosion due to the nature of the project. The access road, constructed to facilitate removal of the dam, served as a barrier to prevent the silt, previously deposited in the reservoir, from moving downstream.

In addition, to help reduce the erosion problem, the dewatered reservoir area was seeded aerially with a helicopter.

Demolition of the Dam

The demolition of the timber component of Mussers Dam was completed on November 21, 1992, culminating a two month construction effort.

All demolition work was performed by American Hydro Power Company with a subcontract to their parent company, Conduit and Foundation, Inc. Laborers were hired locally and worked under the supervision of their own foremen. Generally, the construction crew included the foremen, plus 1 or 2 equipment operators, and 4 to 6 laborers. In addition, a consulting engineer was retained as construction manager.

Heavy equipment used on site included a D-4 bulldozer, a track-mounted backhoe, a Caterpillar 953 track-mounted front end loader, a Grove 35 ton hydraulic crane, several dump trucks, and several flat-bed trucks.

Care of Water

The reservoir draw down process commenced on September 25, 1992, following removal of the hydraulic turbine, diverting flow through the powerhouse. To avoid unnecessarily high flows, a steel plate bulkhead was placed over the vertical draft tube opening, and a hole was cut in the plate equivalent to the 54-inch runner diameter. This limited the flow to the 250 cfs hydraulic capacity of the turbine.

An initial discharge of silt was noted when the intake gates were opened, but within the first 15 minutes of dewatering, the inflow into the turbine pit cleared. In addition, concerns regarding scouring in the draft tube and tailrace were alleviated when field observations showed no sign of scouring of the foundation.

On Tuesday Sept. 29th the orifice plate and a 3-foot by 14-foot section of the timber flooring in the turbine pit were removed to accelerate the draw down process. In addition, two 3-foot wide by 4-foot high openings were cut in the face of the timber dam at the waterline in the vicinity of the existing diversion gates used during the original construction of the dam.

The reservoir water levels were completely drawn down by Monday morning, Oct. 5, 1992, and a flow of approximately 20 cfs was passing through the opening in the dam where the boards had been removed.

Dewatering of the reservoir revealed the existence of an old timber crib dam about 25 feet upstream of Mussers Dam. This structure reportedly dates back to the mid 1800's when a mill operated on the site. The water wheel was located within a 16-foot wide sluiceway, located near the center of the timber crib dam. The diversion gates in the upstream face of Mussers Dam were located just downstream of this sluiceway, which carried the entire flow of Middle Creek during most of the construction period.

Dewatering efforts continued through the remainder of the week of October 5th as the construction crews removed a three foot high swath of face planks from the upstream face of the dam along the silt line. The bottom four planks on the downstream face were also removed across the

full length of the dam. This provided a sufficient opening to pass storm flows in excess of 25 percent of the 100 year event.

Efforts to lower the downstream water surface included removal of some of the debris in the downstream channel, and removal of about 20 feet of a stone crib wall on the right side of the powerhouse tailrace channel. This resulted in reducing the depth of water from 25 inches to 15 inches above the dam's concrete foundation slab.

The timber crib dam upstream of Mussers Dam was widened to serve as an access road, working platform, cofferdam and silt barrier. Five 20-foot long sections of 24-inch diameter steel pipe were placed in the sluiceway to carry river flows.

The timber crib working platform was washed out twice during construction when it was overtopped by high flows, once on November 3rd, and again during the weekend of November 14th. After the second failure, four additional 30-inch diameter pipes were added. On November 22nd, the day after demolition of Mussers Dam was completed and the pipes were removed, flows in Middle Creek reached about 1,000 cfs. This flow was well in excess of any flows previously experienced during the demolition period.

Access Road Construction

Anticipating the use of the upstream timber crib structure as a platform for the crane, an access road was constructed from the boat ramp on the left bank of the reservoir, along the left shoreline, and then along the upstream toe of the earthfill section of the dam.

A small bulldozer was initially used to excavate the silt down to rock along the access road. Unfortunately, because of the extremely flat angle of repose of the silt material, the bulldozer was unable to push the silt from the powerhouse intake channel area. Silt in this area was about six feet deep.

The solution to this problem was to create a silt barrier with imported granular material. Silt was excavated from in front of the leading edge of the access road with a track mounted backhoe, and granular material was dumped in its place before the silt could slide back into the excavation.

By October 19th, the access road had progressed onto the timber crib dam to a point about 80 feet to the right of the powerhouse. Further work on the access road was

postponed for a short time while the backhoe excavated the silt from the toe of the dam along that 80-foot length.

A total of approximately 2,500 cubic yards of granular material was imported to construct the access road. Some of this material was used to widen the crest of the timber crib dam so that it could be used as a working platform for the hydraulic crane and the backhoe. As discussed above, on two occasions this working platform was washed out where it crossed Middle Creek and had to be rebuilt.

Dam Removal

The demolition of Mussers Dam commenced on September 29, 1992 with the removal of the first few face planks. Since the face planks were of a tongue and groove design, a chain saw was needed to cut the tongue of the first plank. Then the remaining planks were removed by lifting with pry bars. Generally two men worked together on each plank. Because of the weight of the creosote impregnated 3x10 planks, two men were needed to carry each one to the stockpile. The old timber crib dam was used for the location of the stockpile. The planks were stacked in bundles equivalent to about one cord each, banded, and loaded on flatbed trucks for removal from the site.

Initially, two 3-foot wide by 4-foot high openings were made at the waterline in the vicinity of the existing diversion gates on the non-overflow section. By October 5th planks on the upstream face had been removed from the silt level upward to a height of three feet, across the full length of the dam. The remaining planks, both upstream and downstream, were removed by October 26th, with the exception of the planks located below the silt level on the spillway section of the dam.

Removal of the triangular shaped structural timber bents began on October 26, 1992. The 72 bents, spaced at 3'-7" on center, were made up of 8" x 8" structural members diagonally cross-braced with 2x10's. In addition, the bents were tied together with a grid of horizontal 2x10's located at the intersections of the diagonal cross-braces and the vertical 8x8's.

Conservative procedures were implemented in removing the bents. Safety, salvage of useable materials, and the potential loss of materials to the creek, all played a part in the choice of procedure.

Each bent was dismantled, piece-by-piece. The top 8x8 member that supported the upstream face of the dam was removed first using the hydraulic crane. This usually

took two picks because of the splices in the top member. Often a third pick was required because the beam splintered and broke near the bottom/upstream end.

The near vertical 8x8's were then removed one at a time working from downstream to upstream. The diagonal cross-bracing and the horizontal bracing members were pried loose from each 8x8 prior to pulling it loose with the crane. The vertical members were nailed and doweled to the bottom horizontal 8x8 which was embedded in a concrete slab. The crane was then used to pull loose the 2x10 diagonal cross-braces from their connection with the bottom member.

After three or four bents were removed, the horizontal bracing that tied those bents together were removed and lifted to the stockpile with the crane. As was done with the face planks, all timbers and bracing members were carefully stacked by type in one-cord bundles, banded, and removed from the site.

The removal of the bents proceeded at a rate of about six bents per day. This gradually accelerated to about eight bents per day during removal of the somewhat smaller spillway bents. The last bent was removed on Saturday, November 21, 1992.

Visual observations showed that most of the top 8x8 timbers parallel to and supporting the upstream face were deteriorated at their upstream ends, some severely so. The degree of deterioration was accentuated during removal of these members. In many cases the 8x8 timbers splintered and broke in a zone about 10 to 20 feet from the upstream end.

Some of the short vertical 8x8 members near the upstream end of the dam were also found to be deteriorated, and the downstream face planks of the spillway section were in poor condition.

Conversely, most of the face planks on the dam were in very good condition, as were the longer vertical timbers. The face planks were saturated with creosote. The 8x8 timbers were impregnated with creosote to a depth of only one inch. This undoubtedly led to the poor condition of some of these members.

Most of the face planks and bracing lumber was donated to local farmers who hired local flat bed trucks to transport the materials to their farms. Several flat bed trailer loads of the timbers were sold to a used lumber supplier in Brooklyn, New York, while the Pennsylvania Fish and Boat Commission took 60 timbers for

their use at another site. The smaller pieces and rotted timbers were removed from the site by a local carting company in 6 or 7 dumpster loads. Approximately 200,000 board feet of lumber was removed from the dam.

The original turbine was donated to the Agricultural and Industrial Museum of York County in York, Pennsylvania.

Water Leakage through Cracks in Reinforced Concrete

Donald O. Dusenberry[1], Member, ASCE,
and Steven J. DelloRusso[2]

Abstract

Leakage of water through reinforced concrete is an issue of interest to professionals in water power, water supply, and sanitary water industries, as well as engineers involved in floating and offshore concrete structures. Researchers have studied the factors that affect water flow between plates, and some investigators have studied flow rates through cracks in rock and unreinforced concrete; however, there is little published on flow of water through natural cracks in reinforced concrete elements. This paper summarizes tests of water flow rate through a cracked reinforced concrete element.

Introduction

For smooth, planar cracks with parallel surfaces, flow follows a cubic law that can be derived from the Navier Stokes equation for laminar flow of incompressible viscous fluids. The expression for flow rate is:

$$Q = T \frac{dH}{d\ell} \tag{1}$$

where

$$T = \frac{g\, b^3}{12\, \nu} \tag{2}$$

[1]Senior Associate, Simpson Gumpertz & Heger Inc., 297 Broadway, Arlington, Massachusetts, 02174

[2]Senior Engineer, Simpson Gumpertz & Heger Inc., 297 Broadway, Arlington, Massachusetts, 02174

in which T is transmissivity, $dH/d\ell$ is the hydraulic gradient through the thickness of the element, g is acceleration due to gravity, b is the crack width, and ν is the kinematic viscosity of water.

The practical application of design codes for reinforced concrete structures usually results in cracks that are relatively small (typically less than approximately 0.5 mm) at the surface of concrete elements. In structures with cracks of this width, flow is dependent on factors that are not included in Equation 1: e.g. crack surface roughness, tortuosity, constrictions in the flow path, and duration of flow.

In effect, surface roughness creates boundary layers in which flow velocity is diminished. In fine cracks, the effective width of these boundary layers can be a significant fraction of the width of the crack. Tortuosity and constrictions in the flow path change what would otherwise be one dimensional flow into two dimensional flow, and reduce the effective aperture of the crack. In portland cement concrete, autogenous healing can close the effective opening of fine cracks when flow rates are small. The simple cubic relationship for transmissivity does not account for these factors, and therefore overestimates the flow rate through fine cracks.

This paper summarizes a series of tests on a 203.2-cm (8-inch) thick concrete test specimen that was reinforced to model the conditions in an existing structural element. Transmissivity adjustment factors derived from these tests are appropriate for estimation of flow through the modeled element, and may be applicable in general to reinforced concrete elements with similar geometric and hydraulic characteristics.

Literature Review

Most of the previous researchers have proposed factors to adjust transmissivity to account for the factors that affect flow in cracks through concrete and rock. The adjusted flow rate equation is:

$$Q = C \cdot T \frac{dH}{d\ell} \tag{3}$$

where C is the adjustment factor.

Louis (1969) performed tests of flow between artificially roughened uniform surfaces with separations, or apertures, ranging from 2 mm (0.08 inch) to 25.4 mm (1.00 inch). Based on these tests, he proposed adjust-

ment factors for transmissivity for laminar and turbulent flow. For laminar flow, Louis (1969) proposed the following adjustment factor:

$$C = \frac{1}{\left[1 + N\left(\frac{k}{D_h}\right)^{1.5}\right]} \tag{4}$$

where k is the average asperity height, D_h is the hydraulic diameter (equal to two times the aperture, b, of a crack), and N is an empirically determined coefficient. Louis (1969) established a value of 8.8 for N. Amadei and Illangasekare (1992) attempted to verify this factor recently through laboratory tests using artificially roughened surfaces; they found N = 42. Amadei and Illangasekare (1992) attribute the difference between their coefficient and that of Louis (1969) to differences in the roundness of the sands that were used to artificially roughen the crack surfaces.

Amadei and Illangasekare (1992) developed a different adjustment factor to account for surface roughness. They defined relative roughness as follows:

$$R = \frac{\epsilon}{2b} \tag{5}$$

where ϵ is the difference between the diameters of the largest and smallest sand grains used to roughen the crack surface.

For values of relative roughness less than approximately 0.2, Amadei and Illangasekare (1992) found that their data could be represented by the following adjustment factor:

$$C = (1 - 2.2\ R)^3 \tag{6}$$

For cracks with relatively high relative roughness ($R > 0.2$), Amadei and Illangasekare (1992) found that the adjustment factor, C, maintained a constant value of approximately 0.28.

Variations in the flow path trajectory out of the average crack plane, sometimes called macroroughness or waviness, is one form of tortuosity. These flow path variations affect both the total effective length of the crack and the energy losses at trajectory changes. Researchers (Sharp, 1970; Amadei and Illangasekare, 1992) proposed adjustment factors based on the average deviation angle of the trajectory from the plane of the crack. Results from both studies show that these effects are small for deviation angles less than approximately 15°.

Several researchers (Louis, 1969; Iwai, 1976; Chen et al., 1989) have proposed adjustment factors for obstructions and constrictions in the flow path. Iwai (1976) experimented with smooth parallel cracks by placing obstructions in random patterns in the flow path. He proposed an adjustment factor of:

$$C = \frac{1}{4}\left[\frac{5}{4c+1} - 1\right] \tag{7}$$

where c is the ratio of the area of contact between opposite surfaces of a crack to the total crack surface area. Amadei and Illangasekare (1992) verified this adjustment factor during their recent laboratory studies.

Test Procedures

Whereas this study was undertaken with a specific structure in mind, the depth of the concrete section, the size and placement of reinforcement in the test specimen, and the hydraulic head applied to the cracked surface were all predetermined. The reinforced specimen was cast monolithically, allowed to cure for seven days, and physically cracked using hydraulic jacks mounted in a custom fabricated steel load frame. The hydraulic jacks were then replaced with fine-thread screw jacks for crack width control. A controlled hydraulic head was applied to a premeasured crack and the quantity of water passing through the cracked specimen in a given time period was collected and measured.

The specimen was reinforced with two #8 bars that extended from end to end through the crack zone, and cages of longitudinal and hoop steel confined to each end zone. The #8 bars, which were threaded at the ends to provide a means for attachment to the loading frame, were placed on the centerline of the specimen, one behind the other in the direction of flow, 38 mm from each face of the specimen. All deformed bar reinforcement was ASTM A615 Grade 60 steel. Figure 1 shows the reinforcing steel in the form prior to casting.

The concrete was a 34.48 MPa (5,000 psi) mix with 19 mm (3/4 in.) "Ossipee", or river washed, gravel coarse aggregate. The design specified a water-cement ratio of 0.40, and 3.5 to 6.5% entrained air. Concrete was obtained from a ready-mix supplier and vibrated into the forms. Curing occurred at ambient temperature (18^{o} to 24^{o}C) under wet burlap. Additional unreinforced specimens were cast from the same batch as the test specimen. These specimens were cored and tested for compressive strength in accordance with ASTM C42 seven days after casting, the same day that the test crack was initiated in the test specimen. The average strength of the three cores tested was 33.7 MPa (4,890 psi).

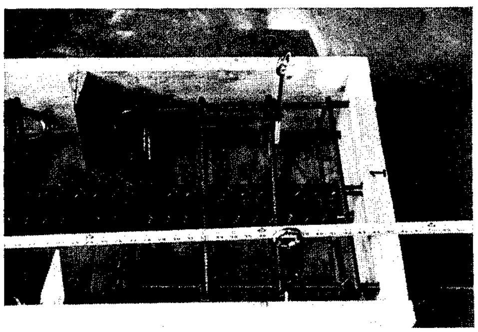


Figure 1. Steel reinforcement in right half of test specimen (wood blocks to create notches are at left)

Notches were cast in the sides of the specimen as stress concentrators to force crack initiation at a predetermined location. A 305 mm (12 inch) net specimen width at the crack location was selected for convenience.

After the crack was initiated, the hydraulic jacks were removed and replaced with screw jacks at the corners of the load frame. The sides of the specimen at the crack location were sealed with butyl ribbon sealer packed tightly into the notched area (the root of the crack) and held mechanically in place with wooden block wedges. The crack opening on the underside of the specimen was framed with a piece of ribbon sealer to act as a drip edge. The test specimen was kept moist as practicable throughout the test period.

The screw jacks were adjusted manually to establish a uniform crack width at the surfaces of the specimen as indicated by measurements through a 50X optical comparator at six locations each on the top and bottom surface of the specimen.

A layer of butyl sealer was affixed to the perimeter of the pressure chamber and the assembly was rigidly clamped to the top of the specimen. Hydraulic head was controlled via an elevated reservoir with a continuous water supply and overflow tube. A schematic of the test configuration is shown in Figure 2.

When the desired test crack width had been established, the test hydraulic head of 2.13 m (7 ft) was applied to the top surface of the specimen and water was allowed to flow through the crack. The flow rate through the crack was measured by collecting water in a tared collection

pan placed underneath the specimen for a measured time period, and weighing the quantity of water collected. The same specimen was utilized for all tests. Between tests, the pressure chamber was removed and the screw jacks were adjusted to establish the next test crack width. Water temperature during tests was approximately 18°C.

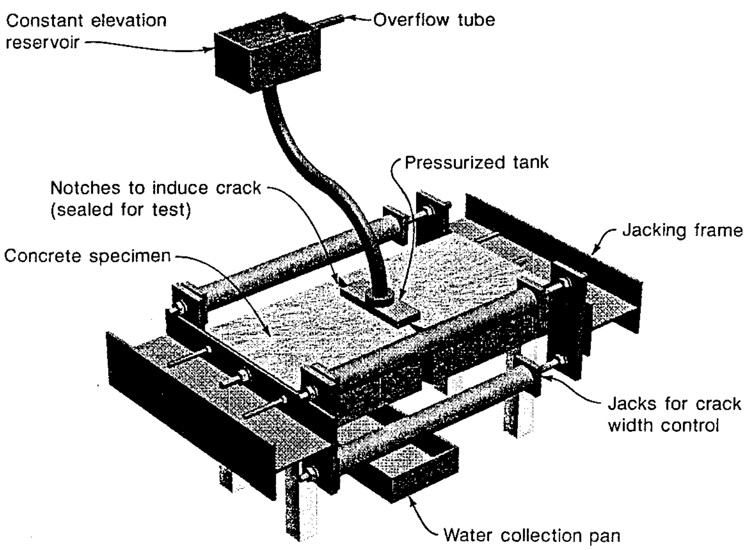


Figure 2. Test configuration

While loading the specimen for Test 3, fine secondary transverse and longitudinal cracks (at the reinforcing steel) formed in the test specimen. These cracks were monitored for water flow and observed to contribute none. Test data are summarized in Table 1.

Results

Based on the work of prior researchers, we postulated the following correction factor for laminar flow transmissivity of the tested natural crack:

$$C = K\left[1 - \frac{b_o}{b}\right]^3 \qquad (8)$$

where b_o represents a threshold crack width below which measurable flow does not occur, and K is a factor to account for tortuosity and blockage.

Table 1 – Summary of Test Results

Test	Average Crack Width (mm)	Test Time (min.)	Flow Rate (g/cm-sec)	Transmissivity mm^2/sec
1A	0.28	30	0.199	1.71
1B		30	0.178	1.53
2	0.09	60	0.003	0.02
3A	0.47	6	1.184	10.17
3B		5	1.058	9.08
3C		5	1.011	8.69
3D		5	0.992	8.52
4A	0.17	30	0.136	1.16
4B		30	0.122	1.05
5A	0.33	15	0.292	2.51
5B		20	0.284	2.44
6A	0.16	30	0.090	0.77
6B		60	0.082	0.70

Figure 3 shows the test data and the corresponding curve with the empirically determined coefficients K = 0.118 and b_o = 0.013 mm. The value of Equation 8 varies between 0.068 and 0.109 for the range of crack widths tested (0.09 to 0.47 mm).

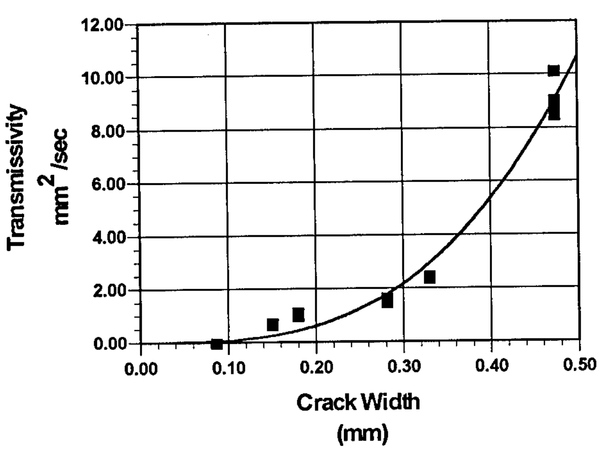


Figure 3. Test data and best fit curve

Discussion

Prior researchers have studied flow of water through cracks in concrete. In order to evaluate the effects of the various parameters that influence flow, they have endeavored to carefully isolate, quantify, and make uniform each of the parameters. Unfortunately, in natural cracks surface roughness and crack width are not uniform over the surfaces of the cracks, and the amount of effective blockage created by crossing steel is not easily quantified (and may vary with crack width). Also, asperities in natural fractures are mated on opposite crack surfaces, not randomly positioned as in specimens with artificially created surface roughness. While some of these differences between prior researchers' test specimens and natural cracks might not be significant for relatively wide cracks, for fine cracks accuracy of the surface modeling is of major significance. The goal of the present study was to investigate flow through natural fine cracks, with sizes that are outside of the range of applicability of available data, in a specific structural element under investigation.

The test specimen was prepared to represent the materials and configuration of the subject structural element. The test crack was induced by loading the specimen in tension. Crack width was controlled to be uniform at the surfaces of the specimen, but not at interior points. The interior contours formed naturally in response to stresses and deformations caused by loading the test specimen.

To compare the findings of this study to the findings of other researchers, one must attempt to quantify the relative roughness of the natural crack. While no specific measurements were made, visual examination of the crack at the surface of the test specimen revealed that the asperity height in the tested cracks, which were all less than 0.5 mm, was a large fraction of the aperture. Relative roughness, R, for these tests was not less than approximately 0.25, and, for very fine cracks, approached the limit of 0.50 as defined by Louis (1969).

Amadei and Illangasekare (1992) report that the transmissivity adjustment factor remains essentially constant for relative roughnesses above approximately 0.2. In fact, all of their data for specimens with large roughness (ϵ = 1.21 mm (0.0477 inches)) is relatively insensitive to relative roughness over the entire range of tests (relative roughness between 0.11 and 0.31). Amadei and Illangasekare (1992) state that the plateau might be related to flow between sand particles (which created asperities that were not well mated on opposite surfaces of the crack) at high relative roughnesses, and that the absolute roughness in these tests was larger than observed in natural fractures of concrete. For these reasons, it is

possible that the plateau observed by Amadei and Illangasekare (1992) does not represent flow characteristics in natural fractures.

Amadei and Illangasekare (1992) do not report data for medium (ϵ = 0.579 mm (0.0228 inches)) and fine (ϵ = 0.292 mm (0.0115 inches)) roughness for values of relative roughness, R, larger than approximately 0.13 and 0.17, respectively, so it can not be determined from their data if adjustment factors based on these specimens would also exhibit plateaus. For relative roughnesses smaller than these values, Amadei and Illangasekare's (1992) data for medium and fine roughnesses are well represented by Equation 6.

If one assumes that a plateau does not exist in natural fractures, then Equation 6 results in adjustment factor magnitudes that continue to decrease with increasing relative roughness. For relative roughnesses in the range of 0.25 to 0.30, the value of Equation 6 is in the same range as determined by Equation 8 based on the present study. For larger values of relative roughness, Equation 6 underestimates flow. The similarity between values of Equations 6 and 8 over a narrow range of relative roughness might be overstated, however, since the present study included blockage from crossing steel that was not included in Amadei and Illangasekare's (1992) tests.

On the other hand, if a plateau does exist in natural fractures, then adjustment factors based on Equation 7 for blockage ratios between 0.25 and 0.38 are sufficient to reduce Amadei and Illangasekare's (1992) plateau value of 0.28 to values consistent with Equation 8. Blockage ratios on this order do not seem unreasonable for the specimen and test conditions of the present study.

During the present study, flow was measured over more than one time interval for most of the crack widths. In each case, flow was greatest in the first interval and diminished in subsequent intervals. For the test series with the largest number of data points (0.47 mm crack), the flow rate diminished by approximately 16% (the largest reduction noted) and appeared to be approaching a constant rate within approximately 1 hour of the initiation of tests at this crack width. While fine cracks experiencing slow percolation of water are known to "heal" due to the leaching of calcium hydroxide from the cement, there was no indication that this autogenous healing was the cause of flow variations during the subject tests. Additional tests are necessary to verify the existence of and establish the cause of time-dependent flow rates in new cracks.

The subject tests were performed for only one hydraulic gradient and water temperature and with a single specimen configuration.

Additional tests are required to verify the applicability of the results to other conditions.

Conclusion

Tests were performed to determine the rate of water flow through new fine cracks in a specific reinforced concrete element. The tests revealed that the theoretical cubic relationship between aperture and transmissivity for smooth plates can be modified with an adjustment factor to estimate flow. The adjustment factor takes the following form:

$$C = K \left(1 - \frac{b_o}{b}\right)^3$$

where K = 0.118 and b_o = 0.013 mm for the tested conditions and b is the crack width in mm. This adjustment factor, which accounts for surface roughness, tortuosity, variations in the crack width, and flow path blockages in the tested specimen, may be used to estimate flow through new cracks smaller than approximately 0.5 mm in some reinforced concrete elements subjected to relatively small hydraulic gradients.

References

Amadei, B., and Illangasekare, T. (1992). "Uplift Pressures in Cracks in Concrete Gravity Dams: Experimental Study." Report to Electric Power Research Institute, Volume 8, 138 pp.

Chen, D. W., Zimmerman, R. W., and Cook, N. G. W. (1989). "The Effect of Contact Area on the Permeability of Fractures." *Rock Mechanics as a Guide for Efficient Utilization of Natural Resources: Proceedings of the 30th U.S. Symposium*, Morgantown, WV, June 19-22, pp. 81–88.

Iwai, K. (1976). "Fundamental Studies of Fluid flow Through a Single Fracture." *Ph.D. Thesis*, University of California, Berkeley, 208 pp.

Louis, C. (1969). "A Study of Groundwater Flow in Jointed Rock and its Influence on the Stability on Rock Masses." *Rock Mechanics Research Report Number 10*, Imperial College, London, England, 90 pp.

Sharp, J.C. (1970). "Fluid Flow Through Fissured Media," *Ph.D. Thesis*, University of London, Imperial College of Science and Technology, London, 181 pp.

HYDRAULIC MACHINERY MODEL TESTING

IMHEF TEST FACILITIES

Prof. P. HENRY
head of IMHEF

Eng. H.-P. MOMBELLI
test facilities manager

1. INTRODUCTION

The **Institute of Hydraulic Machines and Fluid Mechanics (IMHEF)** is part of the Swiss Federal Institute of Technology in Lausanne.

Acceptance tests on the IMHEF hydraulic machine model testing facilities, according to the IEC standards, are an important industrial activity of the IMHEF technical team. In addition, prototype testing in connection with field acceptance tests can also be carried out.

Comparative tests on models are also very useful when the refurbishment of an existing hydroelectric power plant is planned. In fact, the efficiency tests allow an accurate measurement of the efficiency and power increase of the new runner as compared to the one in operation. Thus one can reliably predict the increase in productivity. Comparative cavitation tests provide a very good prediction on the new runner's cavitation behavior on account of a possible correlation between the erosion damage on the prototype and the observed bubbles on the model runner.

Comparative tests between the runners of competing manufacturers are also very useful to obtain objective and real differences in efficiency and power. Given values of efficiency and power at the time of tender are often not very accurate as they are based on estimations which are influenced by commercial pressure.

The accuracy of measurement is of the order of ± 0,2 to 0,3% on account of high quality measuring instruments used.

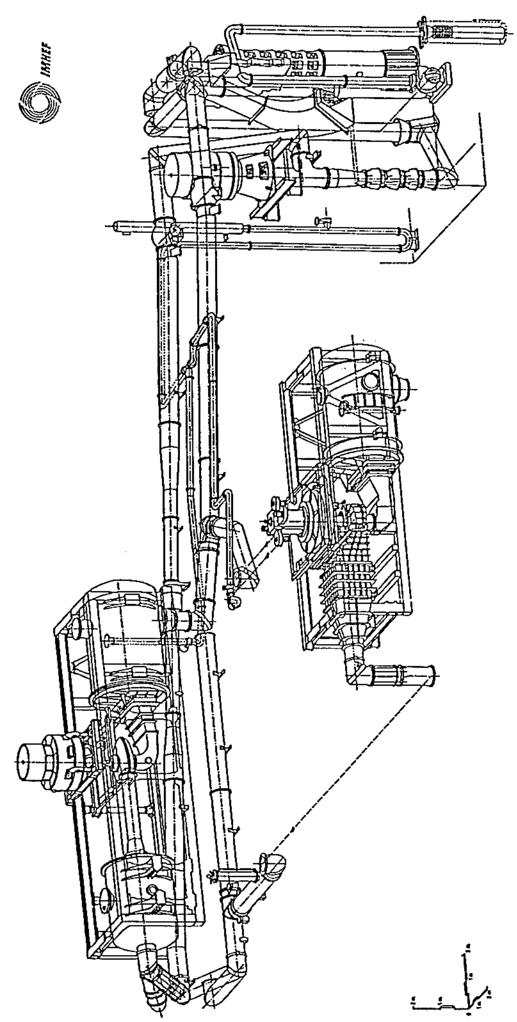


fig. 1 Perspective view of the installation.

2. DESCRIPTION OF THE TEST INSTALLATION

The "Institut de Machines Hydrauliques et de Mécanique des Fluides" (IMHEF) of the "Ecole Polytechnique Fédérale in Lausanne" (EPFL) is equiped with a twin universal test installation for hydraulic machines (see fig. 1), consisting of two different test beds.

Test bed number 1 equiped with a large pressure tank is specially used for Kaplan and bulb type of machines with runner diameters up to 0,5 m. It is also possible to test bulb turbines with a 30 kW generator installed inside the bulb. This solution is very interesting because the homology can be fully respected.

Test bed number 2 is used for Francis turbines, pumps and pump-turbines.

The same control room and therefore the same measuring devices is used for both test beds.

A large free space around the installation makes it possible to install the model with a part or in some cases a complete reproduction of the existing penstock (see fig. 2).

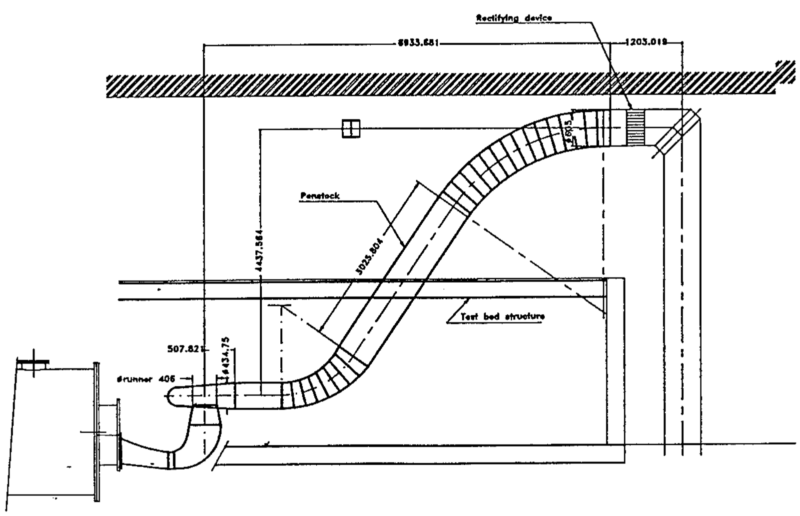


fig. 2 Model reproduced with the existing penstock.

Main characteristics of the test installation are as follows:

Energy:	E	=	20 to 1000 J/kg
Discharge:	Q_{max}	=	1,4 m^3/s
Dynamometer power:	PD	=	300 kW
Pump power:	PP	=	900 kW
Speed:	N_{max}	=	1500 RPM (for test bed 1)
Speed:	N_{max}	=	2500 RPM (for test bed 2)

3. ACCURACY

Accuracy is one of the essential characteristics for a model test installation. IMHEF has been constantly investing time and money in order to update the test installation, by developping and selecting its measuring instruments and techniques for increased accuracy and repeatability.

As a matter of fact, the systematic errors for the efficiency can be written:

$$\frac{\Delta\eta_s}{\eta} = \pm\sqrt{\left(\frac{\Delta E}{E}\right)^2 + \left(\frac{\Delta Q}{Q}\right)^2 + \left(\frac{\Delta T}{T}\right)^2}$$

with:

$\frac{\Delta Q}{Q}$: uncertainty on the discharge. A very large calibration tank and a stable electromagnetic flowmeter guarantee a maximum systematic uncertainty of ± 0,16%.

$\frac{\Delta E}{E}$: uncertainty on the energy. The energy is measured by a load cell connected to a rotating piston manometer. The calibration is made using a dead weight pressure generator. This device gives a systematic uncertainty smaller than ± 0,1%.

$\frac{\Delta T}{T}$: uncertainty on the torque. Two load cells are located on the stator circumference, mounted itself on a hydrostatic thrust-block. The torque measurement is in fact a measurement of load. As for the head this device gives a systematic uncertainty smaller than ± 0,1%.

We thus obtain

$\frac{\Delta\eta_s}{\eta} = \pm 0,19\%$ in the best case and ± 0,26% for very low heads machines.

$\frac{\Delta Q}{Q}$; $\frac{\Delta E}{E}$ and $\frac{\Delta T}{T}$ are directly fonction of the calibration method and of the measuring devices. During the full comparative test the same measurement system is used, thus reducing the uncertainty of the comparison.

Random uncertainties must be limited so that the efficiency uncertainty does not exceed ± 0,10%.

The total uncertainty for efficiency is therefore

$$\frac{\Delta\eta}{\eta} = \pm\sqrt{\left(\frac{\Delta\eta_s}{\eta}\right)^2 + \left(\frac{\Delta\eta_r}{\eta}\right)^2}$$

With the preceding values we obtain an uncertainty between ± 0,21% to ± 0,28%.

In a time when the competition between the manufacturers becomes very hard it is important to guarantee a high degree of precision.

4. TESTS TO BE DONE

4.1. Efficiency

Efficiencies given by manufacturers in their tender documents are very often estimated, are based on similar machines or interpolated between two or more machines of different specific speeds. These estimations are very difficult due to the geometry of the machine which is in most cases not familiar to all the competitors and very often old and not optimum according to modern criteria.

On the other hand, the strong commercial competition entices the manufacturers to guarantee efficiencies sometimes higher than what they are really able to obtain !

Due to these reasons, the customer's choice could be misled due to the poor accuracy of the values of guaranteed efficiencies.

The model test gives the value of the efficiency for the chosen model or the difference between 2 or more manufacturers with a very good accuracy, and helps the customer avoid misinterpretations in the evaluation of different tenders.

4.2. Cavitation

As cavitation results are often very much dependent on the test installation and on the quality of test circuit water (nuclei and dissolved air contents) the results given in the tender are difficult to compare. Comparative test of several runners performed with the same model, in the same test installation and under the same test conditions (head, flow, σ, ... etc.) will give a good and accurate comparison on the cavitation behavior of different runners.

4.2.1. Nuclei influence on caviatation tests

As it is well known and reported for many years, free gas content is important for the case of travelling bubble cavitation inception. Since 1961, Holl and Wislicenius started that the volumetric content of critical cavitation nuclei should be λ^3 time the content of the field, λ being the prototype to model lenght scale. However many experiments carried out at IMHEF for different Francis turbines confirm a strong influence of cavitation nuclei content combined with the test head on the efficiency alteration phenomenon by cavitation. Nuclei content does not only influence cavitation inception but also the development of bubble travelling cavities.

4.2.2. Nuclei saturation effect

The cavitation curves are obtained according to the usual cavitation tests. In addition, air microbubbles are injected in the upstream vessel in order to vary the cavitation nuclei content. It can be observed that for a given threshold value of the nuclei content the alteration of efficiency is no longer affected by increasing the nuclei content, the cavitation coefficient being kept constant.

Moreover, the efficiency alteration is strongly related to the cavitation extent on the blade as it is confirmed by flow visualization. One can observe that the performance alteration is mainly due to the vaporization of a part of the blade to blade channel region which is under the vapor pressure. The saturation phenomenon occurs when the active nuclei amount is large enough to occupy all this region

The strong relation found between cavity extent and efficiency alteration allows us to suggest a modified similitude law for nuclei content based on the total volume occupied by travelling bubble cavitation instead of the number of active nuclei. According to this similitude law, the active nuclei content ratio between the water of the test rig and the water in the prototype turbine should be $\lambda^3 \alpha^{-3/2}$ if α is the specific energy ratio between the prototype and the model. In addition, the similitude should take into account that the lower radius limits of an active nucleus is scaled by the specific hydraulic energy. Owing to the fact that the saturation state is obtained by injecting cavitation nuclei in the test rig, it is possible to write the necessary condition for saturation in the prototype case. By assuming a n^{th} power law distribution for the radii of the cavitation nuclei.

In consequence, if the saturation curve is reached on the model for a typical nuclei content of $2\ 10^6$ nuclei/m^3, the corresponding content at the prototype scale has to be of the order of $0.003\ 10^6$ nuclei/m^3. On site measurements, confirms that this very low value is always exeeded in the water circulating in the water plant.

4.2.3. IMHEF propose a new test procedure for Francis turbines cavitation tests in complement to the IEC standard cavitation tests

Cavitation test conditions

Degassed water	To minimize air bubble problems in the pressure lines and to ensure a good quality of the flow visualization.
Test specific hydraulic energy E	As close as possible to the Froude hydraulic specific energy.
Nuclei injection	To reach the model cavitation saturation curve, which corresponds to the cavitation characteristic of the full-scale Francis turbine.
Measurement of the active nuclei distribution	To control test water quality and cavitation inception value, σ_b
Visual observations	To determine the actual type of cavitation and to evaluate the cavitation erosion risk.
σ level reference	Using the sigma definition given in the IEC standard, with the reference level at the runner outlet edge.

4.3. Pressure fluctuations tests

Pressure fluctuations generated by Francis runners in part-load or full-load operation may widely differ from one runner design to another, even though the rest of the machine is not changed. Besides, specific boundary conditions imposed on the machine by the model test installation may influence the amplitude of dynamic phenomenon. That is why in the case of a competitive test it is better to investigate the dynamic behavior of all runners on the same test loop, using the same test procedure.

4.3.1. Background

Hydraulic fluctuations (of flow velocity and pressure) are a normal feature of Francis turbines operation. They originate in the interactions between rotating velocity fields and non-moving structures of the machine. They are influenced by operating conditions, by the test installation's dynamic response (hydraulic and mechanical), and by the turbine's hydraulic design.

Low frequency fluctuations (below two to three times the runner rotational frequency) are typical of Francis turbines operation. They are basically the part load precession and the free oscillations of the water plug in the draft tube. If the prototype dynamic response is unfavorable, these may be detrimental to the plant stability of operation.

High frequency fluctuations (cavitation noise, wakes) may induce important local forces. As they occur beyond the acoustic cutoff frequency of the connecting pipes, they are normally not hazardous to stability of operation of the turbine, and so they are left out of this investigation.

Generally speaking, model tests provide a characterization of the dynamic behavior associated with a particular turbine design and a range of operation. It provides information essential for a prediction of plant stability: types of observed disturbances, excitation frequencies and acoustic powers as well as the compliance of draft tube cavitation.

4.3.2. Test conditions

Within possibilities of the test installation, the test hydraulic specific energy is chosen so that oscillation amplitudes and frequencies fall within favorable ranges of the instruments. If Froude head is beyond reach or would impose uncomfortable measurement, a more favorable test head is chosen. Care must then be taken in setting the reference elevation for Thoma number σ at the runner outlet outer edge ($\sigma_{\bar{1}e}$).

Minimum testing is a load acceptance at the main plant energy number under a constant test head, at plant σ. Twenty to forty test points are spread from 50% of best efficiency flow, when part load rotation isn't quite organized, to the prototype maximum gate opening, and a little bit beyond that if possible. This amount of data is absolutely necessary for a good diagnosis of the turbine dynamic behavior. Just the test points selected for cavitation observations don't allow a proper perception of dynamic phenomena.

As plant σ depends strongly on the particular conditions of the project, the load acceptance should be repeated at the reference σ, $\sigma_{ref} = 0.2/\psi_{\bar{1}e\ opt}$, which allows comparison between machines of different designs and specific speeds.

For projects with big variations of head, partial tests should also be performed for different energy numbers, with corresponding σ values. It's best to add detailed testing at part load (about 60% of φ_{opt}) and at full load (about 120% of φ_{opt}). These consist in one variation of model rotational speed with constant test head and σ, one variation of σ and possibly a variation of test head.

4.3.3. Instrumentation

Pressure fluctuations are measured using highly sensitive transducers with a suitable dynamic response. The sensors are flush mounted with the turbine model walls.
The tested model is equipped with:

• one pressure sensor on the draft tube cone wall, close to the runner outlet, in the turbine's main axis, on the draft tube outlet side: "downstream cone" (1);
• one pressure sensor on the draft tube cone wall, close to the runner outlet, in the turbine's main axis, on the opposite side: "upstream cone" (2);
• one pressure sensor on the feed pipe wall, at the spiral case inlet: "spiral case inlet" (3);
• if possible, two additional pressure sensors on the feed pipe walls, forming two equal lengths of uniform pipe for acoustic power analysis (4), (5)
• if possible, a torque sensor on the turbine shaft (6)
• according to possibilities, additional pressure sensors on the draft tube wall.

Amplifier output signals are low-pass filtered to eliminate high frequency noise and sampled so as to preserve the phase shift information. The sampling rate is high enough to minimize aliasing effects. Processing is done over at least 400 runner revolutions.
Draft tube cavitation is observed and schematically drawn for each test point.

4.3.4. Particular requirements for pressure fluctuations tests

All wet parts of the model (except seals) must be geometrically similar to those of the prototype. The model construction must be stiff enough to prevent excessive deformations.
The test installation operates in closed circuit mode, so the dissolved gas content can be kept low. There mustn't be travelling bubbles at the model inlet.

4.3.5. Analysis of results

Hydraulic fluctuations associated with Francis turbines operation occur at various frequencies, not multiples of the rotational frequency. The different frequency components of signals must be tracked across variations of operation parameters. That's why recordings are systematically analyzed in the frequency domain. Interpretation of observations is done on the grounds of frequencies, amplitudes and phase shifts for the different operating conditions.

4.4. Runaway speed

Usually, the runaway speed of the existing runner is not known with good accuracy.

The comparison between the runaway speed of the existing runner and the new one gives an accurate and safe value of the difference.

4.5. Mechanical tests

For this kind of tests, the test installation and the model have far less influence and the results obtained in different laboratories are generaly good for comparison purpose.

Nevertheless, tests made in the same laboratory in the same conditions and same test equipment will give a more accurate comparison.

5. COMPARISON BETWEEN EXISTING and NEW RUNNER and also BETWEEN SEVERAL MANUFACTURES

When the refurbishment of a power plant is planned, it becomes interesting to conduct a comparative model test of the existing runner and the newly designed runners on the same test installation. This allows to have a sound reference, i.e. the old runner behavior, known from the owner's experience. All the measured results of the existing runner model may directly be compared to the prototype operation. In the same way, it is then possible to compare runners from competing manufacturers, and the final performance / price evaluation can then be made on the grounds of actually comparable test data, rather than from expected guaranteed values. The final choice is then much safer, regarding efficiencies as well as cavitation, runaway speed, pressure oscillations, etc...

A comparative test of two or more models must absolutely be done in the same laboratory. It is the only way to increase the accuracy of the comparison.

The efficiency test of the existing runner and the new runner is conventional. The accuracy will be evaluated in para 5. It must be noted that the accuracy of the efficiency difference is even better due to some favourable circumstances:

- same test installation
- same model (details are identical, same roughness)
- same magnitude of measured values (head, flow, torque)
- short laps of time between the measurement with the two runners

As for the step - up problem, the best way is to compare directly the measured model efficiencies without step up. The friction losses are almost the same for the model with two different runners. The efficiency difference is mainly due to the decrease of singular losses, specially incidence losses and peripheral velocity component at the runner outlet.

For the same reasons the comparison of power, cavitation behavior, pressure fluctuations and runaway speed of turbines from different manufacturers is also extremely useful.

The eventual cavitation damages of the existing runner are always well known by the owner. The amount and location of repairs during the runner life is usually perfectly recorded. Nevertheless, in most cases the real cavitation pattern on the model runner is not known or was poorly observed during the first acceptance tests. The observation of cavitation bubble extension on the model of the existing runner is extremely useful, to predict accurately the potential area of damages for the new runner.

The only way to perform such comparison cavitation tests is to have the same conditions as for efficiency test.

HENRY P.: Hydraulic Machinery Model Testing.
The Current State of Technology in Hydraulic Machinery pages 63 - 84.
Gower Technical 1986.

These No 914 par GINDROZ B.: Lois de similitude dans les essais de cavitation des turbines Francis, 1991.

BRAND C., AVELLAN F.: The IMHEF system for cavitation nuclei injection, IAHR Symposium, São Paulo 1992.

JACOB T., PRENAT J.E.: Generation of hydro-acoustic disturbances by a Francis turbine model and dynamic behavior analysis. IAHR Symposium, Belgrade 1990.

JACOB T., PRENAT J.E., VULLIOUD G., LOPEZ ARAGUAS B.: Surging of a 140 MW Francis turbine at high load, analysis and solution. IAHR Symposium, São Paulo 1992.

NODA J.M., ASCE M. and PFLUEGER K.A.: Competitive Model Testing for Replacement Runners at the Wells Hydroelectric Project.
Waterpower, Portland, Oregon 1987.

HENRY P. and MOMBELLI H.P.: The interest of competitive model tests. WaterPower conference on uprating and refurbishing powerplants-III, Innsbruck Austria 1991.

Acoustic Flow Measurements at the Rocky Reach Dam

Rick Birch[1] & David Lemon[1]
(ASCE - non members)

Abstract

ASL Environmental Sciences, under contract to the Public Utility District No. 1 of Chelan County, conducted water flow measurements at the Rocky Reach Dam, on the Columbia River. Flow data were required to calibrate models being used in a downstream migrant fish bypass project. Two acoustic measurement techniques were used: an *Acoustic Doppler Current Profiler (ADCP)* to map the circulation within the forebay and tailrace (Dec. 3-9 1991), and an *Acoustic Scintillation Flowmeter* to measure the flow within the turbine intakes (Jan. 7-16 1992).

1. Acoustic Doppler Flow Measurements

1.1 Magnetic Distortion Near The Powerhouse

During previous flow measurements within 40 feet (12 m) of the face of the powerhouse (Figure 1), it was found that apparent magnetic north was shifted 46° clockwise at surface, and changed about 0.7° counter-clockwise with every foot sub-surface. About 500 feet (150 m) away from the powerhouse the distortion was reduced to near zero. The magnetic field near the face was mapped by attaching a current meter to the face with a rigid frame, then monitoring the compass reading while the instrument/frame was lowered. However further away from the face the magnetic field could not be mapped this way and any current meters which use a magnetic compass for direction reference would not be reliable.

----

[1]Oceanographers with ASL Environmental Sciences Inc., 1986 Mills Road, R.R. 2, Sidney, B.C., Canada, V8L 3S1.
Phone: 604-656-0177; Fax: 604-656-2162

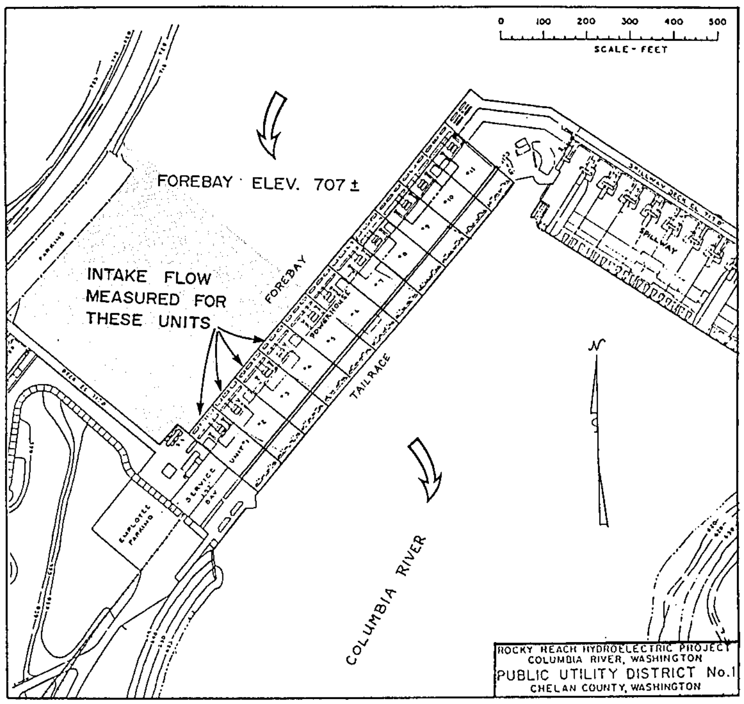


Figure 1. The Rocky Reach hydro-electric dam on the Columbia River near Wenatchee, Washington State. The area of the forebay where ADCP flow measurements were obtained is shaded, while arrows point to units 1, 2, 3, and 4 where intake flows were measured with a scintillation flowmeter.

1.2 Acoustic Doppler Current Profile Flow Measurements

Acoustic Doppler Current Profilers (ADCP) use non-intrusive acoustics to measure flow based on the *Doppler* effect. The instrument can be moored or, as in this case, fixed alongside the boat. Directional information can now be obtained from an external gyro compass, thereby avoiding the problems associated with magnetic compasses. The instrument is submerged until the transducer heads are under water and no reflections are received from the hull. Acoustic pulses are transmitted along four beams (Figure 2). Echoes are returned by scatterers in the water column and the frequency shift of each echo is directly proportional to the component of flow along the beam axis. Horizontal and vertical flow components are then computed from three of the four axial velocities. The fourth is

redundant and provides an estimate of the homogeneity of the flow field. (Directly alongside the powerhouse face one of the beams often reflected off the face; in such cases the software reverted to three-beam solutions.) Echo returns are time-gated to allow flow resolution into vertical bins (3.3 feet, or 1 metre in this case). From surface, the ADCP is thus able to measure the three components of velocity from near-surface to near-bottom at depth increments of order 3.3 feet (1 m).

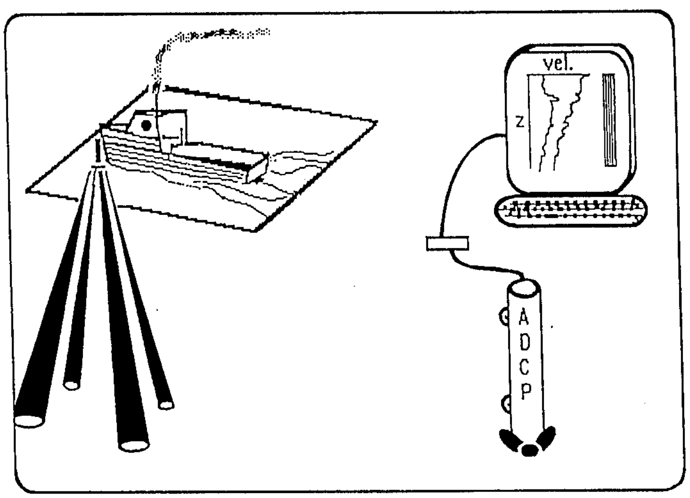


Figure 2. Configuration for the vessel-based Acoustic Doppler Current Profile (ADCP) data collection.

The acoustic beams are directed 30° off vertical therefore the bins from the four beams become more widely separated with depth. The assumption is that the flow field is homogeneous over the area encompassed by the beams. Also, no data are available within about 15 percent of the bottom because of echo returns from the side lobes (sound energy not aligned along the beam axis). Neither are data available within the first few feet of the instrument since the transducers must be allowed to stop ringing before returns can be listened for, and the first bin is often bad. In this set-up the first bin represented a depth of 8.1 to 11.4 feet (2.5 to 3.5 m).

Echo returns off the bottom enable the ADCP to compute the velocity of the boat/instrument to within about ± 0.02 feet per second (fps) (0.006 m/s). This is automatically subtracted from the profile data to produce absolute

current velocities.

ADCP transmission frequency depends on the range required; lower frequencies provide greater range. In this case 1200 kilohertz (KHz) was optimal for the approximately 100 foot (30 m) depths of the forebay. The instrument was of the original "narrow band' configuration produced by RD Instruments of San Diego. Four pings were ensemble averaged to produce velocity data every 1.6 seconds. Given a boat speed of about 5 fps (1.5 m/s), velocity data were available approximately every 8 feet (2.4 m) in the horizontal. The random error of any one ensemble velocity is estimated to be ± 0.22 fps (0.07 m/s).

1.3 Data Collection

Sampling within the forebay was conducted over a 500 by 500 foot (150 by 150 m) area, using a 50 foot (15 m) grid spacing. The vessel was steered along the grid lines and the time on each grid point was used to mate the velocity data to the appropriate grid site. The trash boom caused the survey lines to be disjointed, slowing the survey slightly. A Del Norte micro-wave positioning system was used with three remotes located around the periphery of the forebay. Vessel position was thus known to within ± 1 foot (0.3 m).

Once the ADCP and positioning systems were installed and tested, the actual current survey required less than a day.

Further measurements were made within 50 and 100 feet (15 and 30 m) of the face of the powerhouse with the ADCP attached to the bottom of a 30 foot (9 m) steel pole, which was bolted to the frame at the bow of the boat. The purpose of this was to reduce the sampling footprint of the ADCP in the deepwater near the face, thereby enabling measurements closer to the face and to greater depths.

1.4 Results

For comparison with the physical model results, the ADCP flow measurement data were presented as velocity vectors at each grid site, one plot for each of various depths. The uppermost 11.4 foot (3.5 m) level is shown in Figure 3. The vectors represent the speed (length scaled) and direction of the horizontal component of the flow, with the arrow centred on the grid point. The numbers beside each vector denote the vertical velocity component, positive values indicate an upward flow component.

The flow within the forebay was angled at approximately 45° to the upstream face of the powerhouse. Surface speeds were typically 2 to 3 fps (0.6 to 0.9 m/s). A clockwise eddy exists in the upper left hand corner within which speeds were reduced to about 0.5 to 1.0 fps (0.15 to 0.3 m/s). The flow near the intakes strengthened and rotated counter-clockwise with depth, becoming more perpendicular to the face. The top of the intakes are at a depth of about 50 feet (15 m).

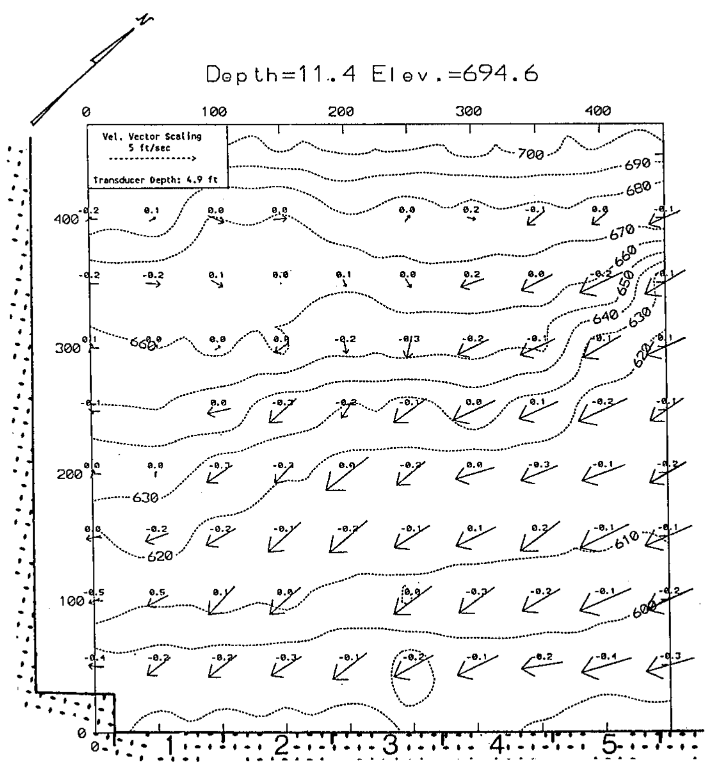


Figure 3. Forebay study area, off units 1 through 5. Contours are elevations in feet. Arrows represent current vectors at 11.4 foot (3.5 m) depth, extracted from ADCP transects at 50-foot (15 m) grid spacing. The vectors indicate the magnitude (length scaled) and direction of the horizontal component of the velocity; the numbers accompanying each vector are the associated vertical velocity components (positive upward).

The flow vectors in vertical sections perpendicular to the

face were also plotted (Figure 4). These also show the strengthening of the flow near the face, as well as the increasing vertical velocity component there.

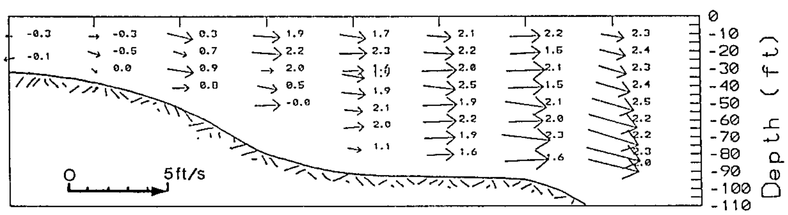


Figure 4. The velocity components at 130° True (perpendicular to the face of the powerhouse off Unit #3, X=250 ft), plotted every 10 feet (3 m) in the vertical. The length of each vector is scaled as to speed, while the accompanying numbers are the component of flow parallel to the upstream face of the powerhouse (at 220° true; positive values indicate flow out of the page).

The results with the ADCP mounted on the pole 30 feet (9 m) sub-surface in water depths of about 100 feet (30 m) near the face were not significantly different than those obtained with the ADCP at surface. This lends support to the assumption of flow homogeneity over horizontal distances of 50 to 100 feet (15 to 30 m), at least in the deeper waters.

2. Acoustic Scintillation Flow Measurements in the Turbine Intakes

2.1 Flow Measurement Using Acoustic Scintillation Drift

Acoustic scintillation drift is a technique for measuring flows in a turbulent medium such as water or air by analyzing the variations (with position and time) of sound which has passed through it. These random fluctuations in the amplitude and phase of the sound are called scintillations and are caused by variations in the refractive index of the medium, which in turn are produced by the turbulence which is part of any real flow. The technique has been used for some time; atmospheric and ionospheric winds have been measured using the scintillation of light or radio waves passing through the atmosphere (Ishimaru, 1978; Lawrence, Ochs & Clifford, 1972; Wang, Ochs & Lawrence, 1981) and flow and turbulence in tidal channels and rivers have been measured using underwater sound (Clifford & Farmer, 1983; Farmer &

Clifford, 1986; Farmer, Clifford & Verrall, 1987; Lemon & Farmer, 1990).

An acoustic scintillation flowmeter measures currents by observing the transverse drift of scintillations between two relatively closely spaced propagation paths (Figure 5). Two transmitters are placed at one end of the paths and two receivers at the other end. No equipment is required within the measurement zone, nor is there any requirement for cabling across it. The system is arranged so that each receiver detects only the signal from the corresponding transmitter. If the paths are sufficiently closely-spaced, the turbulent field does not change significantly during the time required for the current to carry it across the interval between the acoustic paths. The pattern of scintillations observed at the downstream receiver will then be nearly identical to that at the upstream receiver, except for a time lag Δt. The lag Δt may be found by computing the cross-correlation or the covariance of the two signals over some suitable length of record and locating the peak of the curve. In the simplest case, the mean velocity normal to the acoustic paths is found by dividing the path separation by Δt.

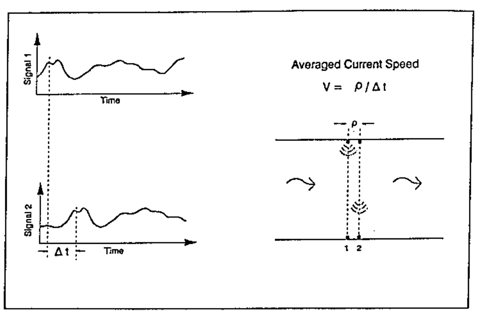


Figure 5. Schematic representation of acoustic scintillation current measurement.

2.2 Measurements at Rocky Reach Dam

Flow measurements were required in the turbine intakes at Rocky Reach Dam for use as reference data in a physical model under construction at Washington State University. The model was to be used in the design of diversion screens to be placed in the intakes to direct salmon fry toward a surface bypass. In this case, the measurements were to be made with no diversion screens in place. The flow information was desired in the form of horizontal averages of the current at a number of levels spanning the height of the intake tunnel.

Measurements were taken in the intake gate slots of turbines 1, 2, 3 and 4. Each turbine intake is divided into three bays (referred to as South, Centre and North). Their location in the dam structure is shown in Figure 1.

The bays are 6.1 m wide and 15.2 m high. The current (averaged across the width of the bay) was measured at 11 elevations as follows (elevations are referenced to geodetic datum at the dam): 634.9, 632.9, 628.9, 624.9, 620.9, 616.9, 612.9, 608.9, 604.9, 600.9 and 592.9 feet.

The flowmeter consisted of two cylinders, each with two transducers mounted at one end and a surface cable at the other. They were mounted on the lower section of the existing fyke net frame which had been shortened and modified by the removal of all the vertical members except the two sides to reduce flow interference. The frame was then positioned at each measurement level and three minutes of acoustic data were collected.

Measurements were made with the transducer pairs oriented both horizontally and vertically, so that the orientation of the flow at each level could be measured. (The geometry of the intake tunnels results in the flow having a significant vertical component which increases with height above the tunnel floor; near the top of the intake, the streamlines are inclined at 45° or more to the horizontal. The generators were operated at nominal full power output of 100 Mwatts during all the data collection runs. The actual generator output and the elevation of the forebay and tailrace waters were recorded for calculating the head during each run to allow the calculated flows to be referenced to a common head.

2.3 Results

The horizontal and vertical components of the flow at each level in each of the bays were calculated from both the acoustic phase and amplitude data. The two sets of data were combined to produce an average. Figure 6 shows the magnitude and direction of the flow in each of the bays of Turbine 4. The length of the arrows is proportional to the flow speed as given by the scale in the upper left of the diagram. The inclination of each arrow shows the direction of the flow at that level.

Figure 7 shows the discharge by bay position for each turbine, both unadjusted for head differences and adjusted to a nominal 27 metre head. The north bays' discharges are consistently lower, being only 75 to 80% of that in the other two bays, for all turbines.

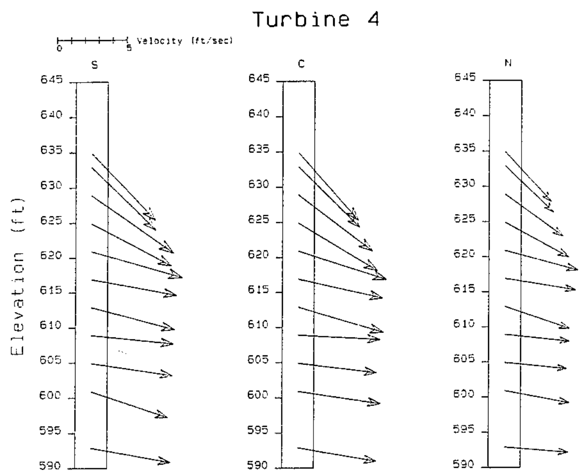


Figure 6. Flow speed and angle for each intake bay.

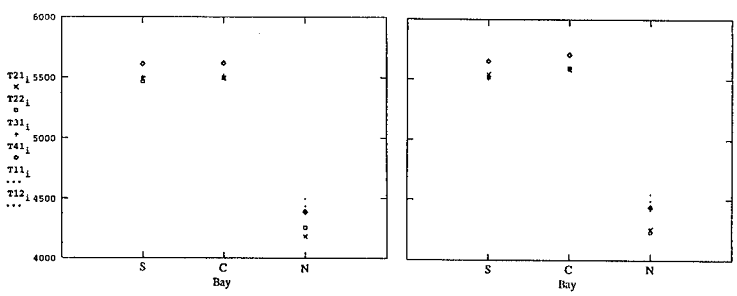


Figure 7. Discharge (in cfs) by bay position for each turbine. T21 = Turbine 2, Run #1, etc. (Unadjusted for head differences: left; adjusted: right)

3. Summary

The Acoustic Doppler Current Profiler (ADCP) proved to be a valuable instrument for obtaining detailed current profile transects in both the forebay and the tailrace of the Rocky Reach Dam. The ADCP was particularly suited to this application because of its ability to operate from surface and use a gyro compass for directional reference. The large variations in the local magnetic field, due to the dam, precluded the use of other more conventional instruments which rely on magnetic compasses.

The acoustic scintillation flowmeter offers a means for rapidly and accurately collecting flow and discharge

information in confined areas such as intake tunnels and penstocks. The measurements can be made with a minimum of flow interference and without the necessity for placing instruments in the measurement zone where they may be subject to damage, or where it may be difficult or expensive to place them without significantly interfering with the flow. The instrument is easily installed, and because it performs an integral measurement, allows spatial averages to be obtained directly, rather than from a series of point measurements. There is no requirement for either a large number of individual current meters, or a time consuming series of multiple instrument placements. Scintillation flowmeters are well-suited to applications such as the one described here, and their use results in significant time and cost savings.

4. Acknowledgements

The authors would like to thank the personnel at Chelan P.U.D. who participated in these studies, in particular Mr. Eldon Rickman who was the engineer in charge.

5. References

Clifford, S.F. and D.M. Farmer, 1983. Ocean flow measurements using acoustic scintillation. J. Acoust. Soc. Amer., **74** (6). 1826-1832.

Farmer, D.M. and S.F. Clifford, 1986. Space-time acoustic scintillation analysis: a new technique for probing ocean flows. IEEE J. Ocean. Eng. OE-11 (1), p. 42 - 50.

Farmer, D.M., S.F. Clifford and J.A. Verral, 1987. Scintillation structure of a turbulent tidal flow. J. Geophys. Res. 92 (C5), p. 5369 - 5382.

Ishimaru, A., 1978. Wave Propagation and Scattering in Random Media. Academic Press, N.Y. 572 p.

Lawrence, R.S., G.R. Ochs and S.F. Clifford, 1972. Use of scintillations to measure average wind across a light beam. Appl. Opt., Vol. 11, p. 239 - 243.

Lemon, D.D. and D.M. Farmer, 1990. Experience with a multi-depth scintillation flowmeter in the Fraser Estuary. Proc. IEEE Fourth Working Conference on Current Measurement, Clinton, MD. April 3-5, 1990. p. 290 - 298.

Wang T.I., G.R. Ochs and R.S. Laurence, 1981. Wind measurements by the temporal cross-correlation of the optical scintillations. Appl. Opt., Vol. 20, p. 4073-4081.

List of Figures

History of Water Power on the Kansas River

David Readio[1]
and
Von Rothenberger[2]

Introduction

The history of water power on the Kansas River can be broken into two distinct eras through three major areas of operation. The first was the nineteenth century, where water power arose as an available source for use in grist mills. Grist mill is an old term for a mill to grind grain[3]. The second era is the twentieth century, when the grist mills were converted into hydroelectric plants.

Rocky Ford Dam

The name Rocky Ford was appropriate for the river crossing on the Blue River seven miles upstream from its junction with the Kansas River near Manhattan, Kansas. A solid bed of rock spanned the stream there, making a shallow place at which to ford the stream during low water. Indians first used Rocky Ford as a river crossing; later a government road was laid out connecting Forts Leavenworth and Riley that crossed the Blue at Rocky Ford.[4]

The first dam was built in 1866 of wooden construction. Huge timbers were bolted to a rock foundation. Low water favored the rapid construction and it was completed December 1, 1866. The dam backed water up the river for eight miles. A sawmill was opened; shingles made there won first prize at the state fair. In 1869, a four-story grist mill was completed from nearby

[1]Plant Operator, Bowersock Mills and Power Co, P.O. Box 66, Lawrence, KS 66044.

[2] History Major, Kansas State University, Manhattan, Kansas.

stone and lumber. Future plans called for a carding factory, cotton factory, and furniture factory, but none of these ever appeared. The pond above the dam became a favorite fishing and swimming hole.[5]

The first dam was swept away by flood waters and a second timber dam was constructed. This second dam was undermined by muskrats in 1877 and the site was then abandoned for over thirty years, until a third dam, with hydroelectric power in mind, was completed in 1910. The new dam was of concrete and masonry construction and extended five hundred feet. There were two large Leffel wheels which could develop 1500 horsepower. This was converted into electricity by Allis-Chalmers generators with 450- kilowatt capacity each. These generators supplied electric power to Manhattan over 6,600-volt transmission lines[6]. In 1913, Rocky Ford was upgraded with a 750-kilowatt turbine and three large new boilers and condensers[7]. The reason for the steam generators was the new 33-kilovolt (KV) transmission line from the Rocky Ford plant to Odgen, Kansas. This line required a regular and unvarying flow of current, which a strictly hydro plant cannot supply due to varying water conditions[8].

Disaster struck in 1919, when the mill was destroyed by a fire started by a bolt of lightning that ignited the roof. Volunteers thought they had control of the blaze, until the cap of a big oil drum was blown off and oil exploded all over the building. The loss was estimated at $100,000 dollars. Interest in the site revived when in July of 1919, it was announced that a $500,000 power plant would be constructed on the east side of the dam. Two months later, a section of the third dam was swept away and talk of a new plant ended. The fourth and present dam was constructed in 1920, just below where the previous dams had been. In 1922, several additional water turbines were installed along with a large 3,125 KVA steam turbine[9]. This dam had a tunnel for carrying fuel and gas lines and had entrances at both ends of the tunnel. In 1936, this tunnel was closed to the public by the state safety director[10].

In 1932, the Kansas Power and Light Company (KPL) purchased the plant. By the 1960s, the plant was used only to meet peaking demand. It continued to be used less and less as KPL's needs outgrew it, until Rocky Ford generated its last kilowatt of electricity on May 12, 1965. KPL later donated the dam and surrounding land to the Kansas Forestry, Fish and Game Commission on October 13, 1967[11]. So ended the saga of the Rocky Ford Dam.

Dam in Topeka

Topeka, Kansas, the state capital, was beginning to grow rather rapidly as the end of the nineteenth century approached. Unfortunately, electric power was expensive, which caused prospective industries to bypass Topeka for some place where power was cheaper. It was decided that Topeka needed to build a dam across the Kansas River in order to have a supply of cheap electricity.

The dam was to be seven hundred seventy-one feet long, and constructed on a concrete base. The dam would be twenty feet high, but to reach bedrock the north side had to be forty-two feet in total height--twenty-two feet below ground. The base was envisioned as twenty-nine feet, four inches wide, and the top nearly six feet wide. The upstream face was to have a slope towards the top of 0.0521 inches per foot, with the downstream side to be a curve with a radius of 17.1 degrees. It was estimated that this dam would supply 4,000 horsepower[12]. The raceway was to be one and a half mile in length. The raceway was to be sixty feet wide and twenty feet deep at its source, and fourteen feet deep at its mouth. The raceway would have a cement floor to prevent water seepage[13]. Construction on this dam commenced in western Topeka, 3.5 miles above what is now the Union Pacific Railroad bridge. As construction continued during 1891, the unthinkable happened. A second company wanted to build a dam in eastern Topeka. Also, the Missouri River Commission wanted to use the Kansas River for navigation. There was division within the city council as to which company to buy its power from. All this confusion led to neither dam being constructed, even though the first dam's base had already been completed[14]. Topeka never again attempted to develop any form of hydroelectric power.

Bowersock Dam

The Bowersock Dam is located in Lawrence, Kansas, and is named after Justin DeWitt Bowersock, who was finally able to tame the Kansas River with a frame dam he built in 1874. A frame dam rests on a foundation of trees and rocks. For this dam, the trees for the bottom were at least sixty feet in length. The trees were laid with the tops upstream as close together as possible, and each end fastened together by means of a two-inch plank spiked on top as each was laid in, the tops interlocking so as to form a solid mat of brush. It required five courses, or platforms, to complete the foundation; each successive platform moved five feet further upstream. Also, each platform had to be thoroughly weighted and filled to form a solid concrete structure[15].

Unfortunately, the river was not easily tamed. Three times the dam washed out in the 1870s and early 1880s. J. D. Bowersock then built a fourth dam in 1884. In that year, 300 horsepower of mechanical energy was produced. This energy was used in the milling operations. In 1886, a small hydroelectric turbine was installed. The mechanical energy was distributed through tunnels along a rawhide belt in which businesses attached pulleys to. These businesses were then billed based on the size of the pulley. In 1888, an ice flow broke loose and headed for the dam. It took out one hundred feet of the dam along with turbine wheels, shafts, pulleys, belts, and cables. Some of the equipment remained attached to the cables and was pulled into the river from the second story of the mill building[16].

In 1903, the first of the turbine/generators was installed. This was a 480 volt, 250-kilowatt generator manufactured by the Minnesota Machine Company. The other six generators were 2300 volt generators manufactured by the General Electric Company. The turbines were manufactured by the James Leffel Company, and the last major replacement on them occurred in the 1940s. The equipment in this plant is currently 1950s vintage, but it is still functional. The plant currently has a maximum operating capacity of 2.4 megawatts. The concrete part of the dam stands eight feet high, and wooden flashboards are used to raise the level an additional five feet. There are eighty-three flashboards constructed of five 8′ x 1′ x 2" planks of oak that span six hundred sixty-four feet of the dam. The spillway, located on the north side of the river, is approximately eighty feet in length. The entrance to the forebay is sixty feet. The total width of the Kansas River at the dam site is eight hundred four feet. The maximum water usage by the seven turbines is 2100 cubic feet per second (cfs) at a twenty foot head. There are times during the summer when the plant is not generating due to high water, which results in a net head of less than ten feet. The plant is currently owned by the Bowersock Mills and Power Company and remains a profitable business in the Lawrence area[17].

Notes:

3. Webster's Ninth New College Dictionary, 1983, p. 537.

4. "The Rocky Ford Power Company Has Harnessed The Blue River," Manhattan Nationalist, 1910, p. 11.

5. Ibid.

6. Rich Dalton, "Rocky Ford Dam and Power Plant," Manhattan Mercury-Chronicle, July 18, 1948, p. 47.

7. "Rocky Ford Improvements," Riley County Democrat, April 11, 1913.

8. "Present Electric Power Plant Outgrowth of an Early-Day Grist Mill," Manhattan Mercury - Diamond Jubilee Edition, Oct. 5, 1929, Sec. 4, pp. 3,6.

9. Ibid.

10. Dalton, p. 47.

11. "KPL Gives a Dam," Magazine of KPL, November 1967, no. 11, vol. 24, pp. 4-6.

12. "The Dam," The Daily Capital, Aug. 21, 1890.

13. "Doubt No Longer," Topeka Daily Capital, March 24, 1891.

14. Edward G. Nelson, KPL in Kansas - A History of the Kansas Power and Light Company, 1964, pp. 46-56.

15. "Local News," Lawrence Herald, March 11, 1869.

16. Dale Nimz, **Living With History: A Historic Preservation Plan For Lawrence, Kansas**, 1984, pp. 83-91.

17. Generator/turbine files, Company Records, Bowersock Mills & Power Company, Lawrence, Kansas, 1993.

References:

Bowersock Mills and Power Company Plant Records and Files, Bowersock Mills & Power Company, Lawrence, Kansas, 1993.

Dalton, Rich, "Rocky Ford Dam and Power Plant," Manhattan Mercury-Chronicle, 18 July 1948, p. 47.

"Doubt No Longer," Topeka Daily Capital, 24 Mar. 1891.

"KPL Gives a Dam," Magazine of KPL, Nov. 1967, no.11, vol. 24, pp. 4-6.

Leffel, James and Co. The Construction of Mill Dams, Springfield, Ohio: 1874.

"Local News," Lawrence Herald, 11 Mar. 1896.

Nelson, Edward G. KPL in Kansas - A History of the Kansas Power and Light Company, Topeka, Kansas: 1964.

Nimz, Dale, Living With History: A Historic Preservation Plan For Lawrence, Kansas, Lawrence, Kansas, 1984.

"Present Electric Power Plant Outgrowth of an Early Day Grist Mill," Manhattan Mercury - Diamond Jubilee Edition, 5 Oct. 1929, Sec. 4, pp 3, 6.

"Rocky Ford Improvements," Riley County Democrat, 11 Apr. 1913.

"The Dam," The Daily Capital, 21 Aug. 1890.

"The Rocky Ford Power Company Has Harnessed The Blue River," Manhattan Nationalist, 1910, p.11.

Webster's Ninth New College Dictionary, Springfield, Massachuttes: Merriam Webester Inc, 1983.

SUBJECT INDEX
Page number refers to first page of paper.

AUTHOR INDEX

Page number refers to first page of paper.

WATERPOWER '93

W. David Hall, Editor

Proceedings of the International Conference on Hydropower, Nashville, TN, August 10- 13, 1993

The perceived simplicity of production of hydropower gives it a less dramatic public image than that of some other sources of energy. While hydropower is the most efficient energy technology, opportunities exist to improve its performance and its existence with the environment. This three-volume set surveys up-to-the-minute information and contemporary issues concerning hydroelectric power. Contributors discuss topics ranging from economics and finance to licensing and legal concerns, from dam safety to computer applications. Papers address water quality, diversion screens, and instream flows; turbine impacts on fish and fish protection; the design, manufacturing, and testing of turbines; and the rehabilitation and modernization of various hydroelectric plants, among many other subjects.

9 780872 629240

ISBN 0-87262-924-4